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Iwai et al.

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(54) **SPARK IGNITION ENGINE**

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(73) Assignee: **MAZDA MOTOR CORPORATION**, Hiroshima (JP)

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PCT Pub. Date: **Mar. 13, 2014**

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F02D 41/00 (2006.01)
F02D 41/30 (2006.01)
(Continued)

(52) **U.S. Cl.**
CPC .... ***F02D 41/0057*** (2013.01); ***F02B 11/00*** (2013.01); ***F02B 23/101*** (2013.01);***F02B29/0418*** (2013.01); ***F02D 13/0203*** (2013.01); ***F02D 13/0207*** (2013.01);
(Continued)

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CPC .. F02D 41/00; F02D 41/0057; F02D 41/006; F02D 41/30; F02D 41/3064; F02D 13/02; F02D 13/0203; F02B 11/00; F02M 25/07; F02M 25/0747; Y02T 10/121; Y02T 10/47
USPC 123/294–298, 305, 568.11, 568.12, 123/568.14; 701/103–105, 111
See application file for complete search history.

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Primary Examiner — John Kwon
Assistant Examiner — Jonny H Hoang
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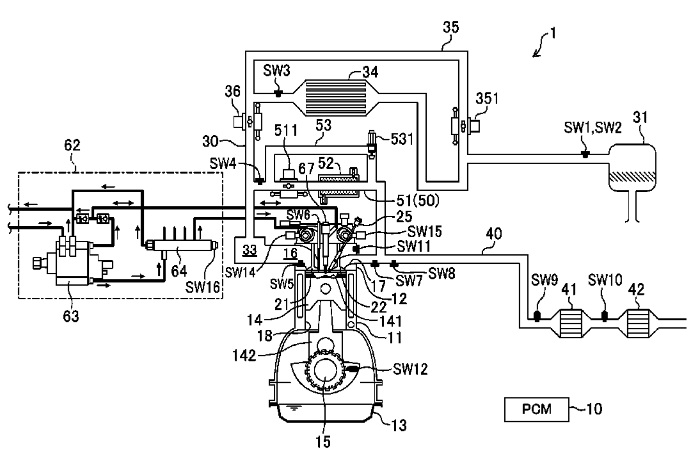
(57) **ABSTRACT**

A controller performs switching between a compression ignition mode in which compression ignition combustion is performed to operate an engine body, and a spark ignition mode in which spark ignition combustion is performed to drive a spark plug to ignite and combust an air-fuel mixture in a cylinder. The controller reduces an EGR ratio to be lower than an EGR ratio set in the compression ignition mode to operate the engine body in a transitional mode in which the compression ignition combustion is performed in switching from the spark ignition mode to the compression ignition mode.

16 Claims, 14 Drawing Sheets



(51) **Int. Cl.**

F02D 21/08	(2006.01)
F02B 23/10	(2006.01)
F02B 29/04	(2006.01)
F02D 13/02	(2006.01)
F02B 11/00	(2006.01)
F02B 1/12	(2006.01)
F02D 41/40	(2006.01)
F02D 37/02	(2006.01)

(52) **U.S. Cl.**
CPC ***F02D 13/0261*** (2013.01); ***F02D 13/0273*** (2013.01); ***F02D 21/08*** (2013.01); ***F02D 41/0002*** (2013.01); ***F02D 41/006*** (2013.01); ***F02D 41/0055*** (2013.01); ***F02D 41/3035*** (2013.01); ***F02D 41/3064*** (2013.01); *F02B 1/12* (2013.01); *F02B 29/0437* (2013.01); *F02D 37/02* (2013.01); *F02D 41/402* (2013.01); *F02D 2041/001* (2013.01); *Y02T 10/125* (2013.01); *Y02T 10/146* (2013.01); *Y02T 10/18* (2013.01); *Y02T 10/42* (2013.01); *Y02T 10/47* (2013.01)

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FIG.1

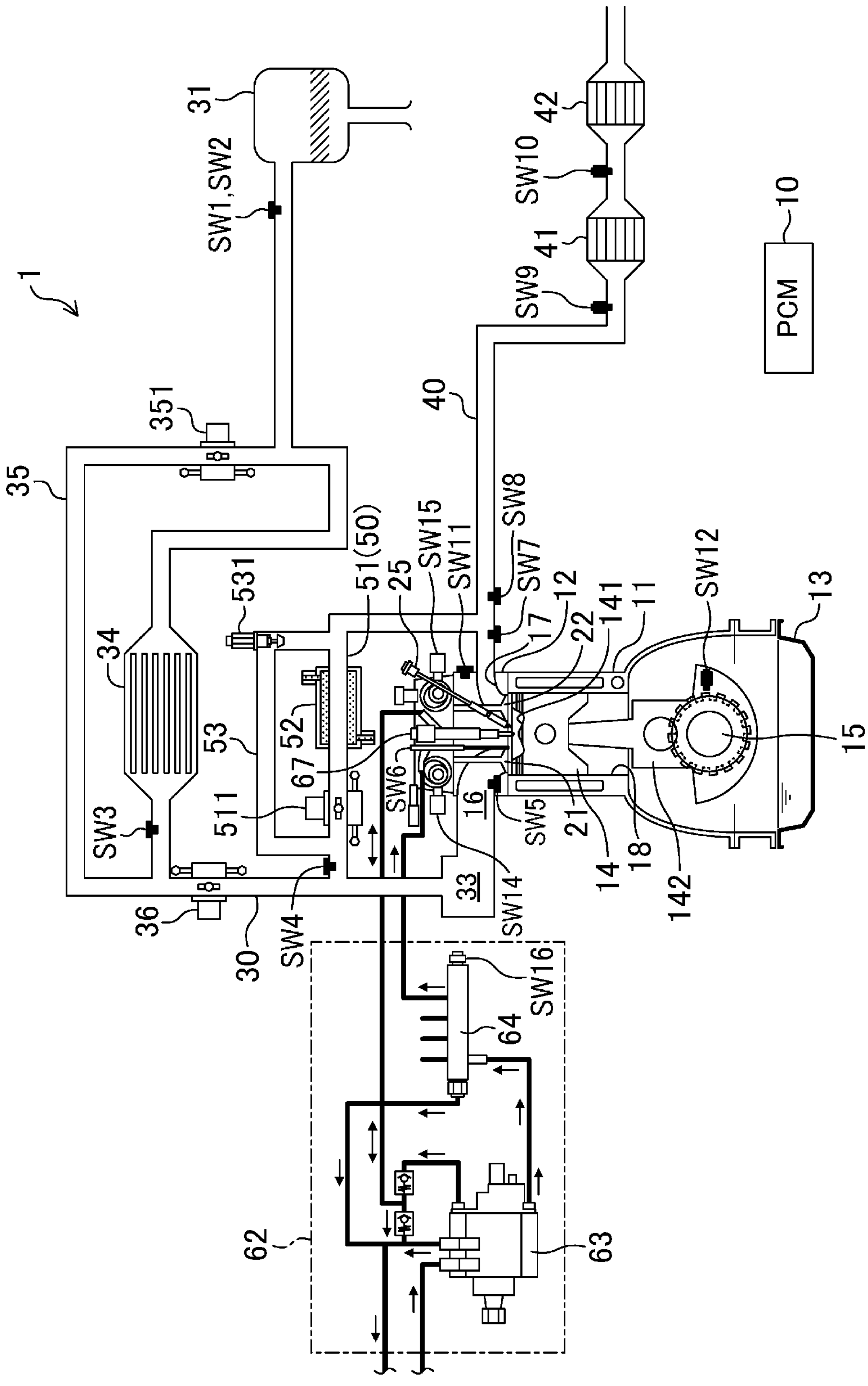

1
31
SW1,SW2
42
SW10
41
SW9
10
PCM
40
351
35
SW8
SW7
12
11
141
22
17
SW11
SW15
25
51(50)
531
34
53
52
67
511
SW3
SW4
36
30
SW6
16
SW5
21
14
18
142
15
SW12
13
33
SW14
SW16
64
62
63

FIG.2

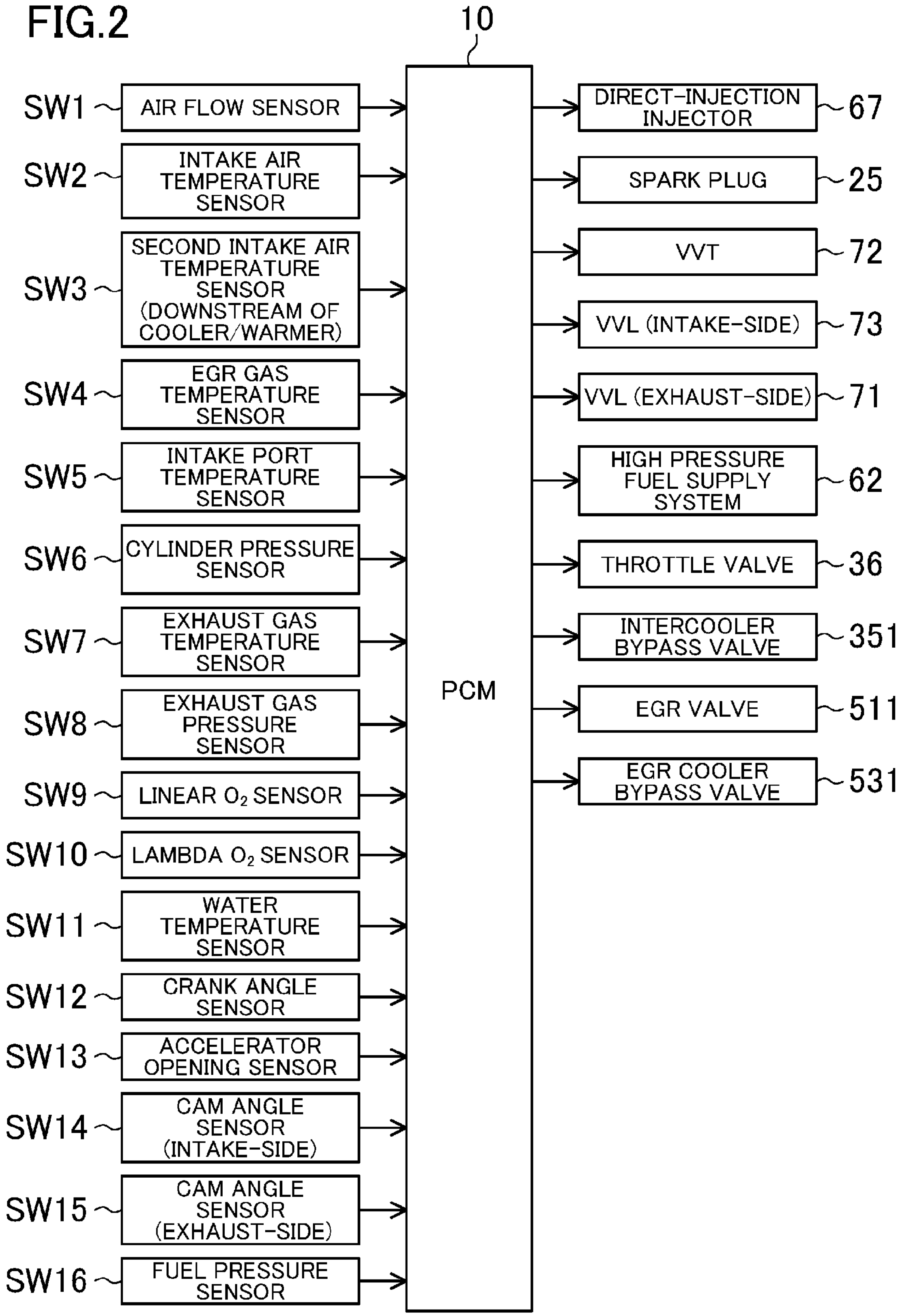

10
SW1
AIR FLOW SENSOR
SW2
INTAKE AIR TEMPERATURE SENSOR
SW3
SECOND INTAKE AIR TEMPERATURE SENSOR (DOWNSTREAM OF COOLER/WARMER)
SW4
EGR GAS TEMPERATURE SENSOR
SW5
INTAKE PORT TEMPERATURE SENSOR
SW6
CYLINDER PRESSURE SENSOR
SW7
EXHAUST GAS TEMPERATURE SENSOR
SW8
EXHAUST GAS PRESSURE SENSOR
SW9
LINEAR O_2 SENSOR
SW10
LAMBDA O_2 SENSOR
SW11
WATER TEMPERATURE SENSOR
SW12
CRANK ANGLE SENSOR
SW13
ACCELERATOR OPENING SENSOR
SW14
CAM ANGLE SENSOR (INTAKE-SIDE)
SW15
CAM ANGLE SENSOR (EXHAUST-SIDE)
SW16
FUEL PRESSURE SENSOR
PCM
DIRECT-INJECTION INJECTOR
67
SPARK PLUG
25
VVT
72
VVL (INTAKE-SIDE)
73
VVL (EXHAUST-SIDE)
71
HIGH PRESSURE FUEL SUPPLY SYSTEM
62
THROTTLE VALVE
36
INTERCOOLER BYPASS VALVE
351
EGR VALVE
511
EGR COOLER BYPASS VALVE
531

FIG.3

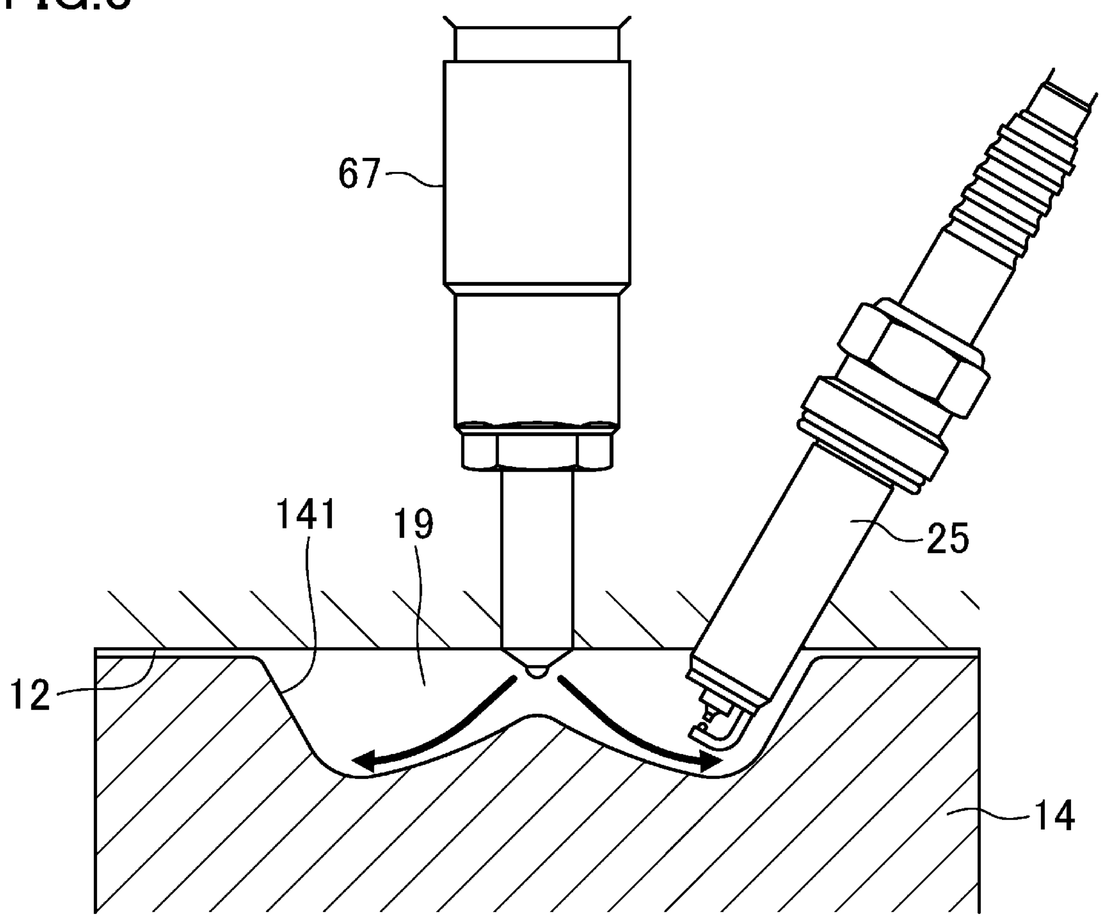
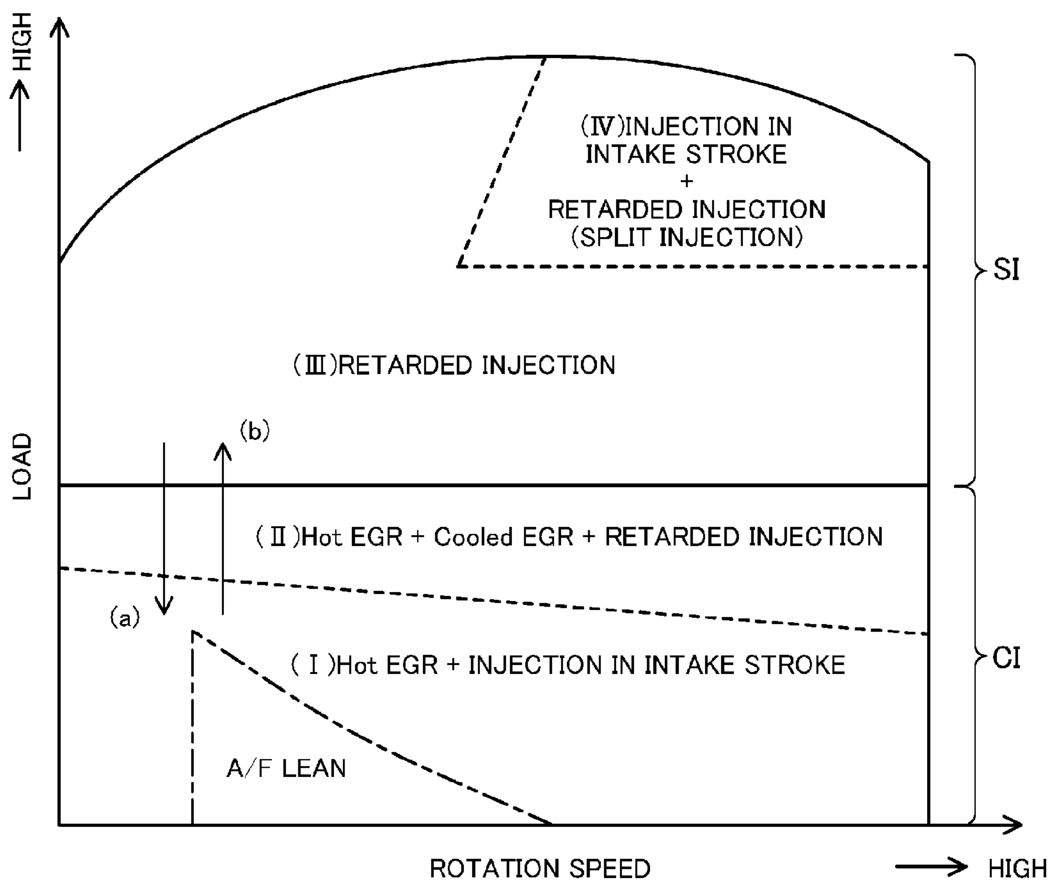


FIG.4



HIGH
LOAD
(IV)INJECTION IN INTAKE STROKE + RETARDED INJECTION (SPLIT INJECTION)
SI
(III)RETARDED INJECTION
(b)
(II)Hot EGR + Cooled EGR + RETARDED INJECTION
(a)
(I)Hot EGR + INJECTION IN INTAKE STROKE
CI
A/F LEAN
ROTATION SPEED
HIGH

FIG.5A

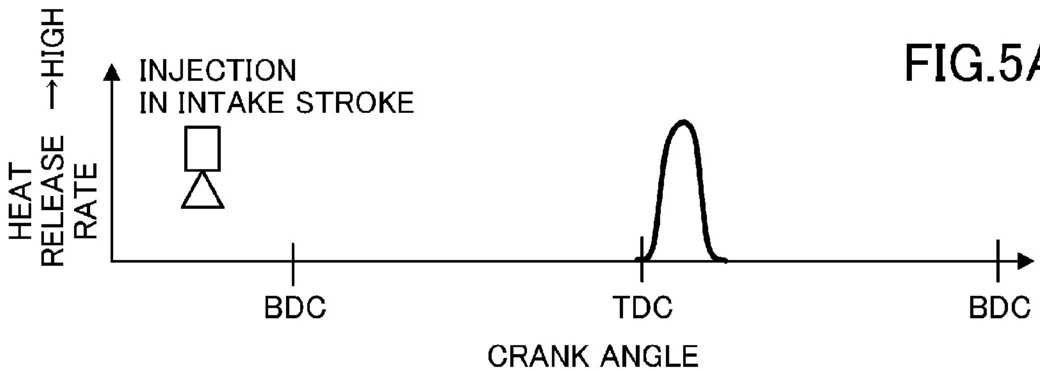


FIG.5B

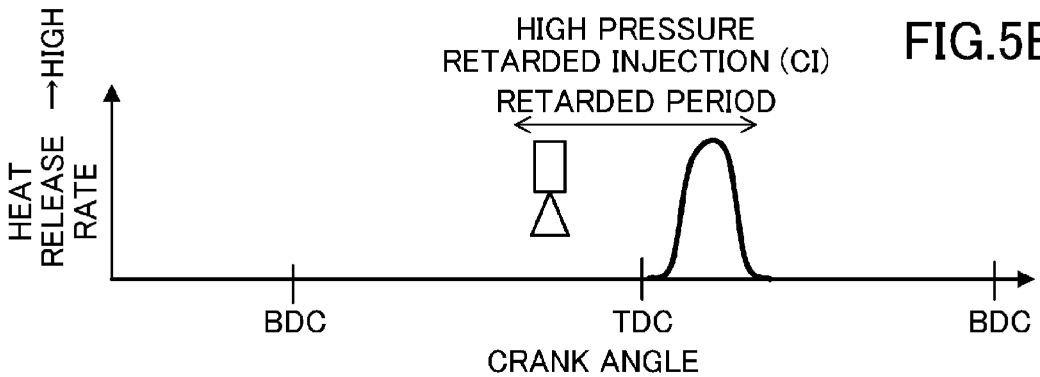


FIG.5C

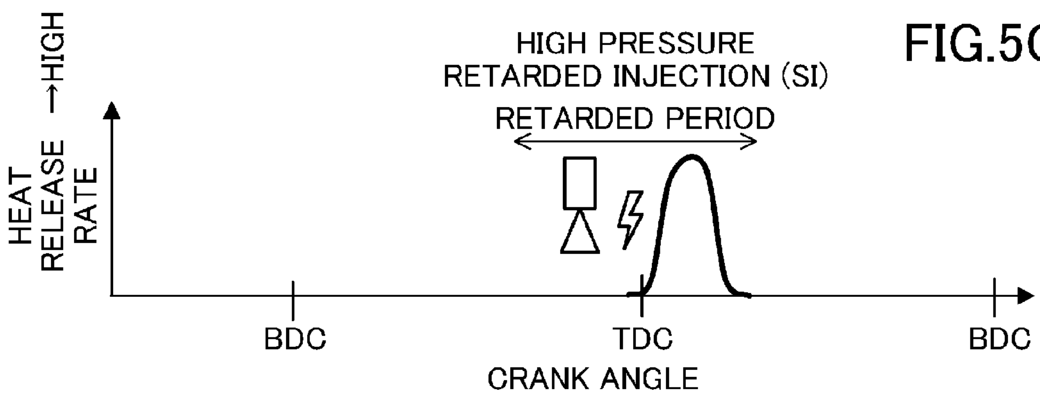


FIG.5D

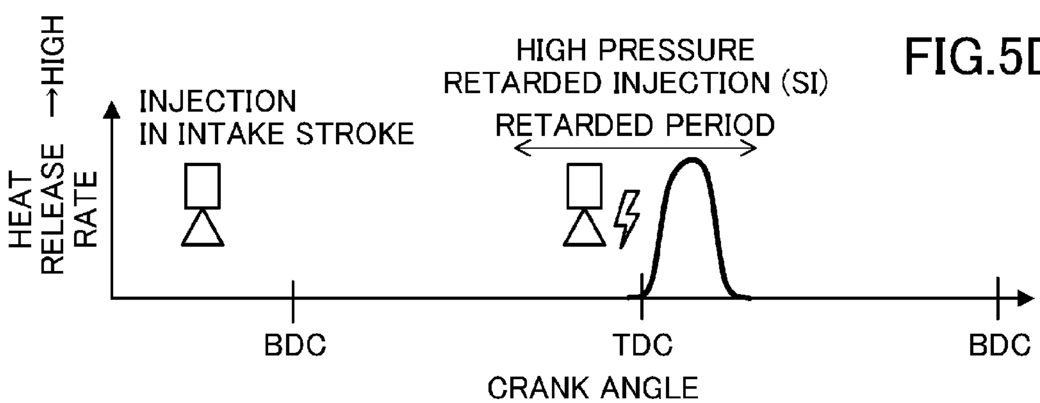


FIG.6

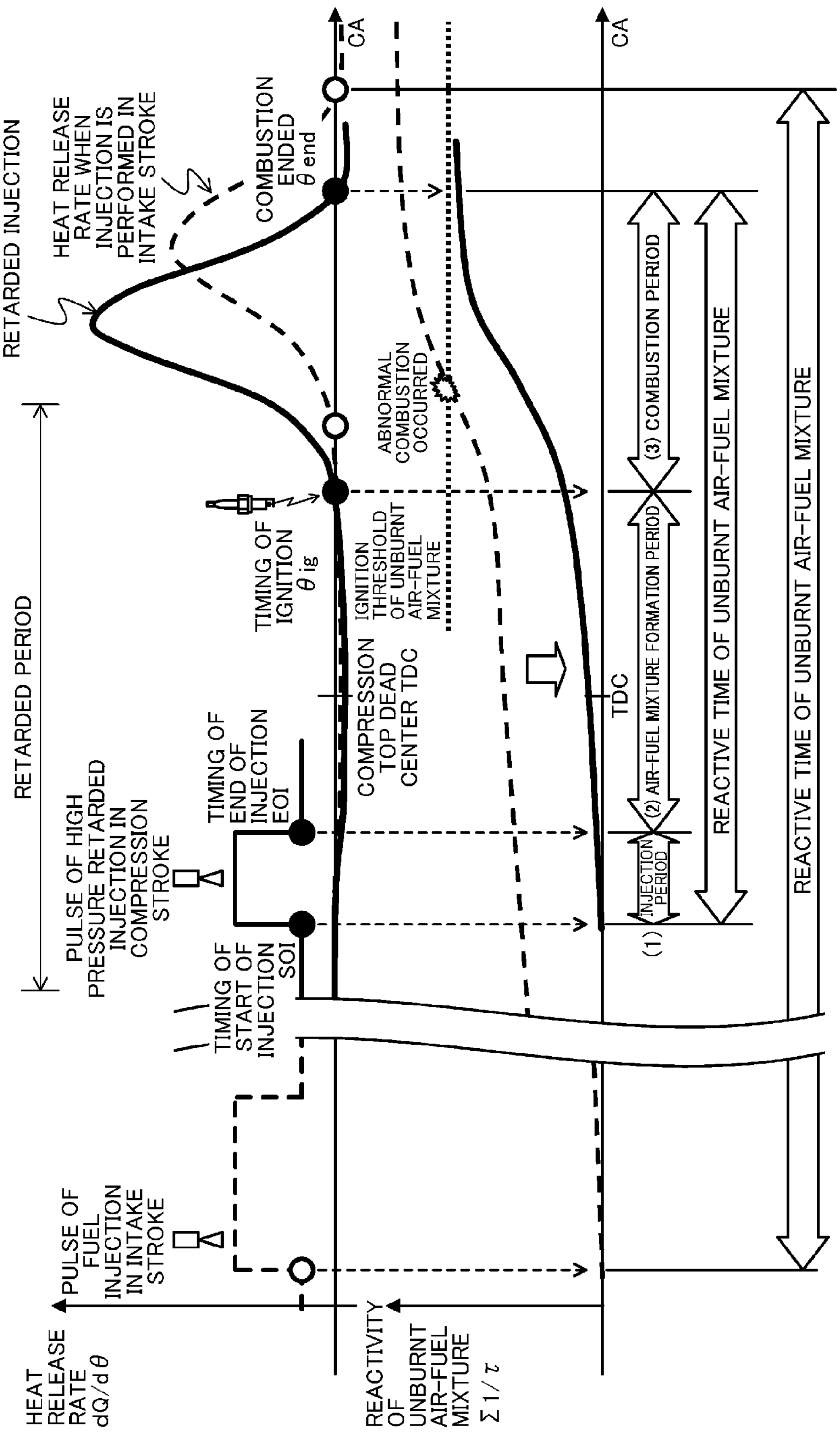

HEAT RELEASE RATE dQ/dθ
REACTIVITY OF UNBURNT AIR-FUEL MIXTURE Σ1/τ
CA
CA
PULSE OF FUEL INJECTION IN INTAKE STROKE
TIMING OF START OF INJECTION SOI
PULSE OF HIGH PRESSURE RETARDED INJECTION IN COMPRESSION STROKE
TIMING OF END OF INJECTION EOI
RETARDED PERIOD
COMPRESSION TOP DEAD CENTER TDC
TIMING OF IGNITION θ ig
IGNITION THRESHOLD OF UNBURNT AIR-FUEL MIXTURE
ABNORMAL COMBUSTION OCCURRED
HEAT RELEASE RATE IN HIGH PRESSURE RETARDED INJECTION
HEAT RELEASE RATE WHEN INJECTION IS PERFORMED IN INTAKE STROKE
COMBUSTION ENDED θ end
TDC
(1) INJECTION PERIOD
(2) AIR-FUEL MIXTURE FORMATION PERIOD
(3) COMBUSTION PERIOD
REACTIVE TIME OF UNBURNT AIR-FUEL MIXTURE
REACTIVE TIME OF UNBURNT AIR-FUEL MIXTURE

FIG.7

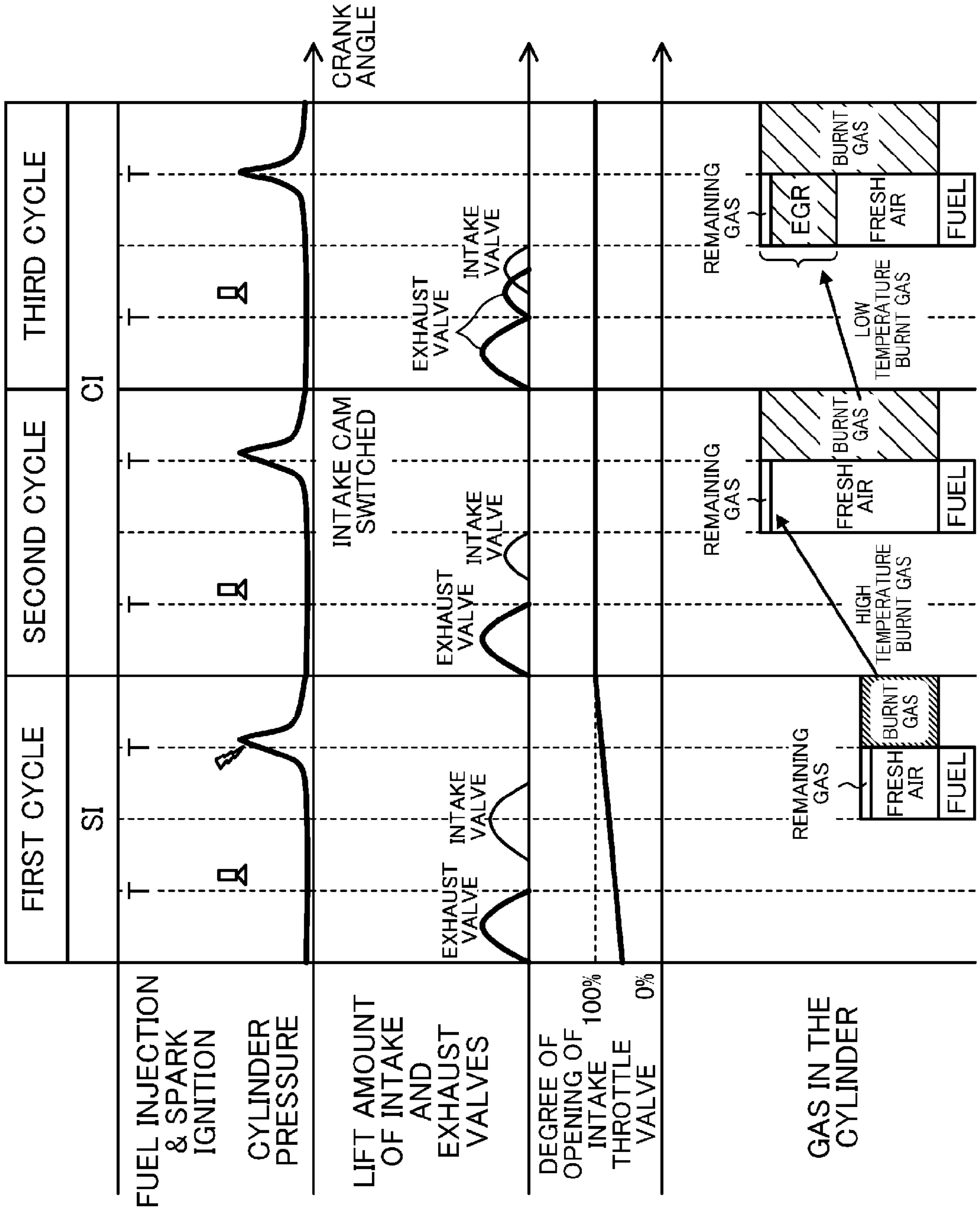

FIRST CYCLE
SECOND CYCLE
THIRD CYCLE
SI
CI
CRANK ANGLE
FUEL INJECTION & SPARK IGNITION
CYLINDER PRESSURE
LIFT AMOUNT OF INTAKE AND EXHAUST VALVES
DEGREE OF OPENING OF INTAKE THROTTLE VALVE
100%
0%
GAS IN THE CYLINDER
T
EXHAUST VALVE
INTAKE VALVE
INTAKE CAM SWITCHED
REMAINING GAS
FRESH AIR
FUEL
BURNT GAS
EGR
HIGH TEMPERATURE BURNT GAS
LOW TEMPERATURE BURNT GAS

FIG.8

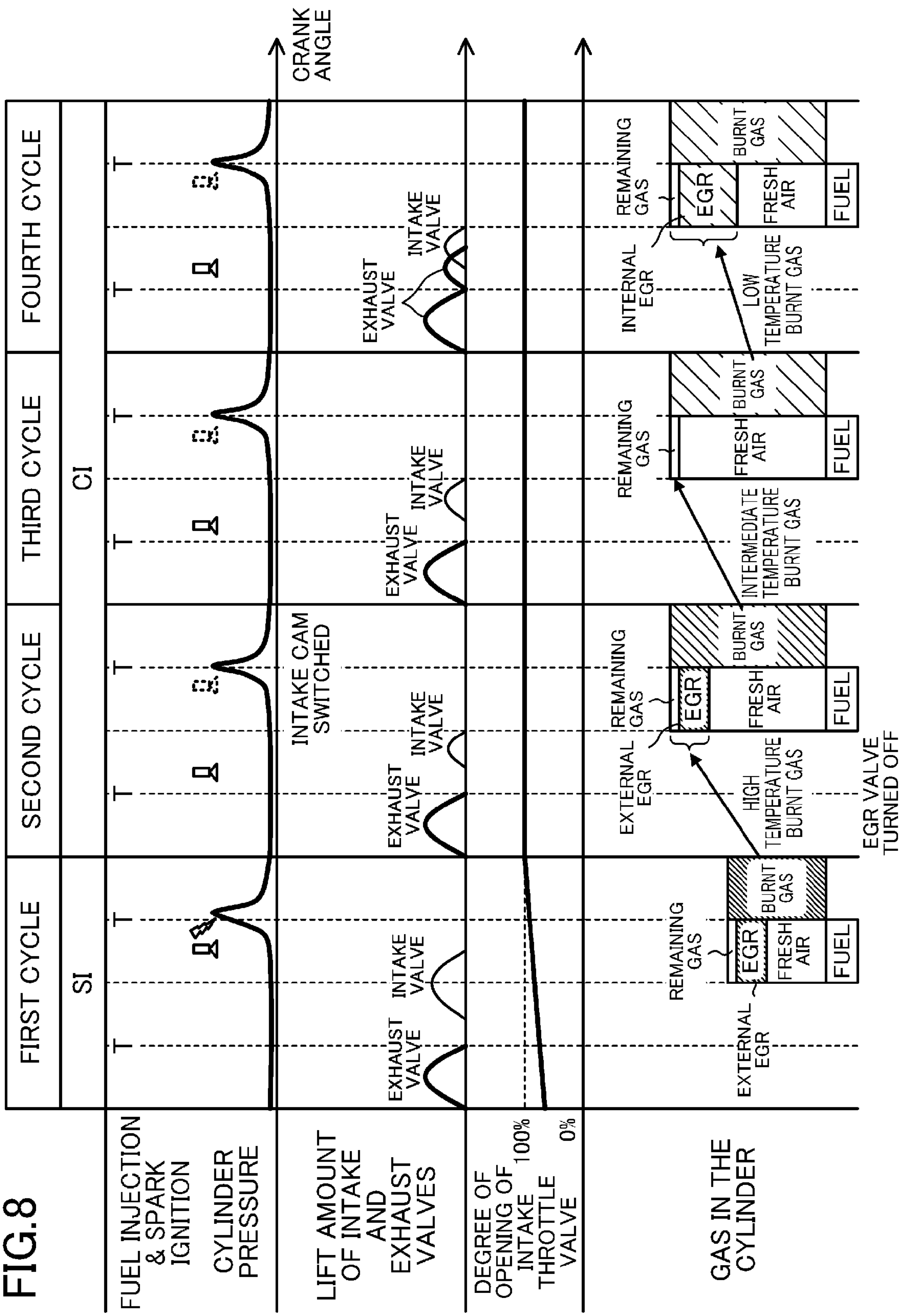
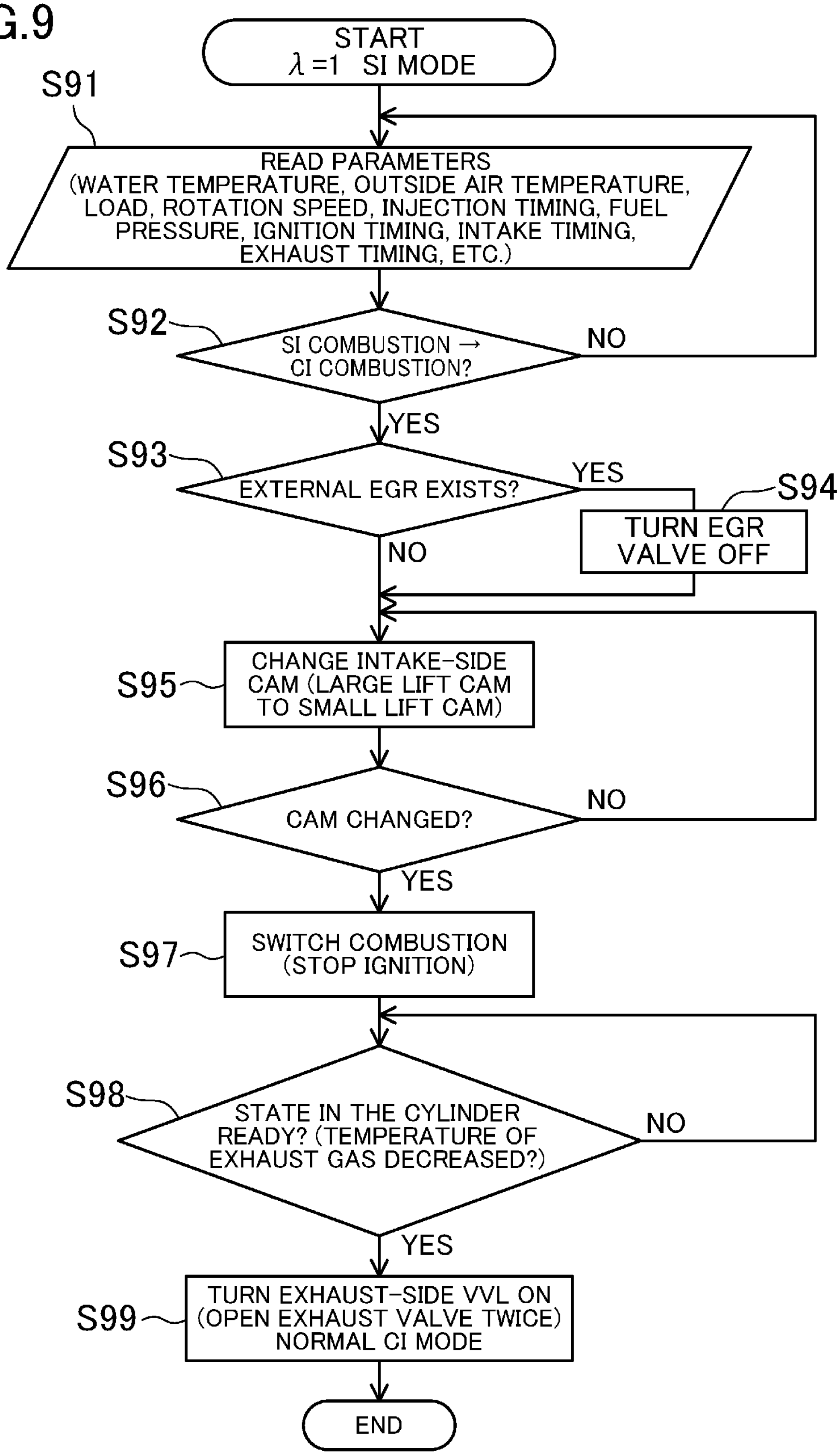

FIRST CYCLE
SECOND CYCLE
THIRD CYCLE
FOURTH CYCLE
SI
CI
FUEL INJECTION & SPARK IGNITION
CYLINDER PRESSURE
CRANK ANGLE
LIFT AMOUNT OF INTAKE AND EXHAUST VALVES
EXHAUST VALVE
INTAKE VALVE
INTAKE CAM SWITCHED
DEGREE OF OPENING OF INTAKE THROTTLE VALVE
100%
0%
GAS IN THE CYLINDER
EXTERNAL EGR
REMAINING GAS
INTERNAL EGR
EGR
FRESH AIR
FUEL
BURNT GAS
HIGH TEMPERATURE BURNT GAS
INTERMEDIATE TEMPERATURE BURNT GAS
LOW TEMPERATURE BURNT GAS
EGR VALVE TURNED OFF
T

FIG.9



START
λ=1 SI MODE
S91
READ PARAMETERS
(WATER TEMPERATURE, OUTSIDE AIR TEMPERATURE, LOAD, ROTATION SPEED, INJECTION TIMING, FUEL PRESSURE, IGNITION TIMING, INTAKE TIMING, EXHAUST TIMING, ETC.)
S92
SI COMBUSTION → CI COMBUSTION?
NO
YES
S93
EXTERNAL EGR EXISTS?
YES
S94
TURN EGR VALVE OFF
NO
S95
CHANGE INTAKE-SIDE CAM (LARGE LIFT CAM TO SMALL LIFT CAM)
S96
CAM CHANGED?
NO
YES
S97
SWITCH COMBUSTION (STOP IGNITION)
S98
STATE IN THE CYLINDER READY? (TEMPERATURE OF EXHAUST GAS DECREASED?)
NO
YES
S99
TURN EXHAUST-SIDE VVL ON (OPEN EXHAUST VALVE TWICE) NORMAL CI MODE
END

FIG.10

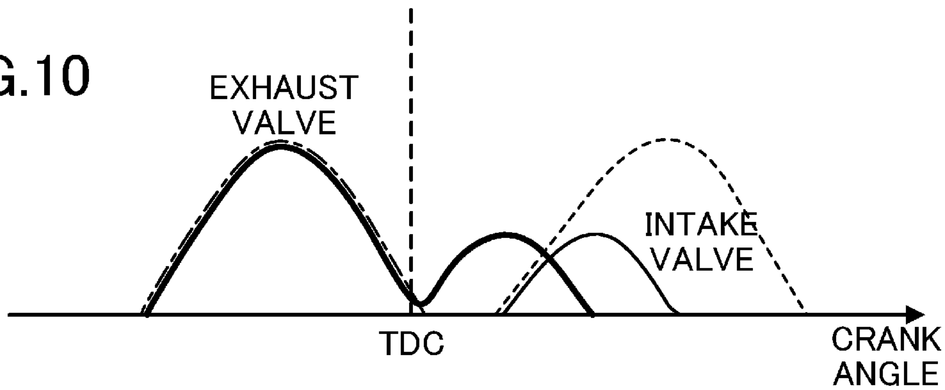


FIG.11

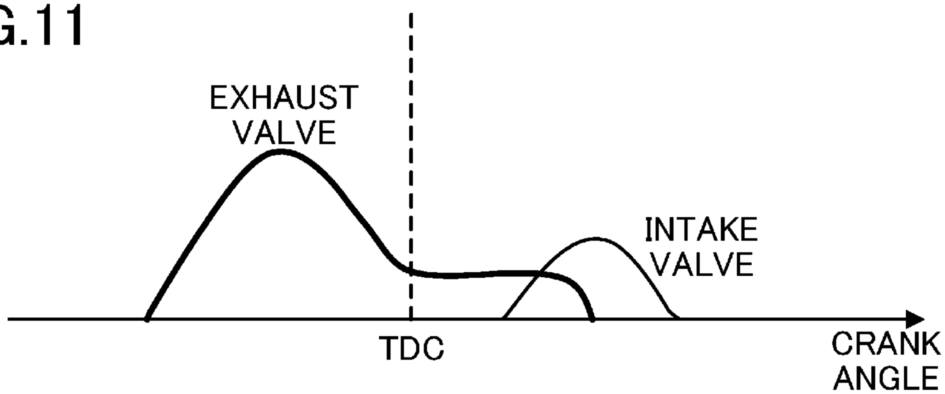


FIG.12

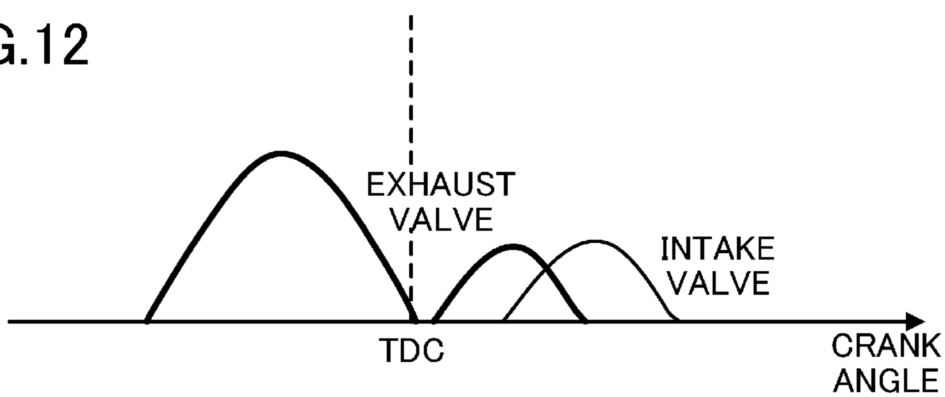


FIG.13

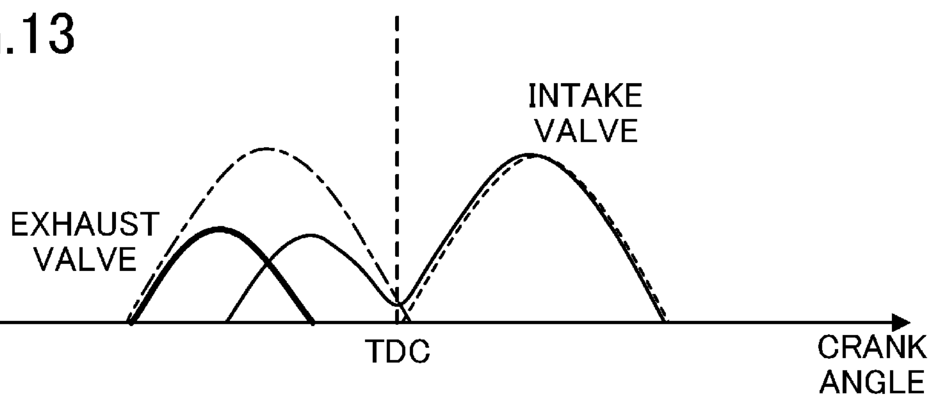


FIG.14

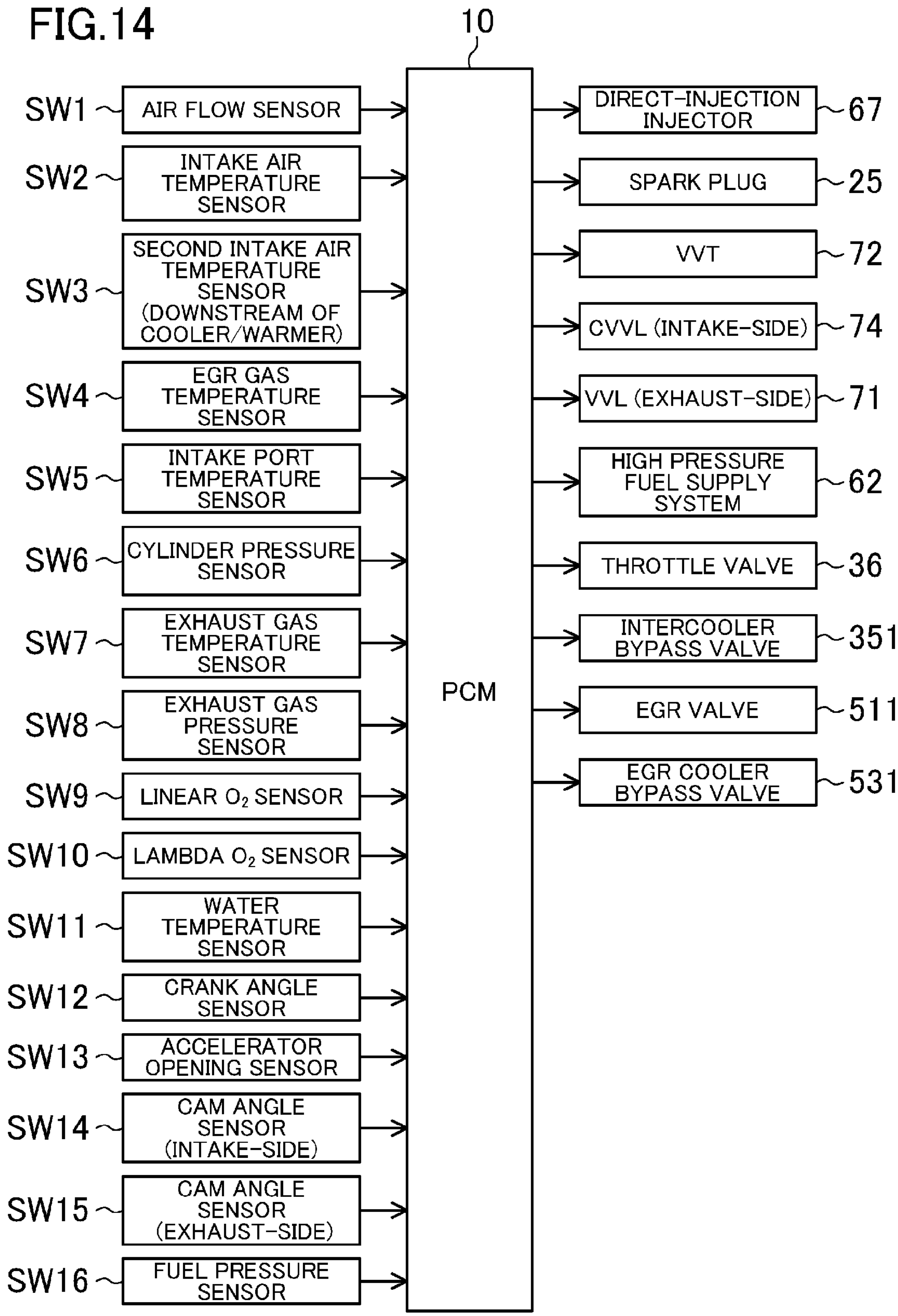

10
PCM
SW1 AIR FLOW SENSOR
SW2 INTAKE AIR TEMPERATURE SENSOR
SW3 SECOND INTAKE AIR TEMPERATURE SENSOR (DOWNSTREAM OF COOLER/WARMER)
SW4 EGR GAS TEMPERATURE SENSOR
SW5 INTAKE PORT TEMPERATURE SENSOR
SW6 CYLINDER PRESSURE SENSOR
SW7 EXHAUST GAS TEMPERATURE SENSOR
SW8 EXHAUST GAS PRESSURE SENSOR
SW9 LINEAR O_2 SENSOR
SW10 LAMBDA O_2 SENSOR
SW11 WATER TEMPERATURE SENSOR
SW12 CRANK ANGLE SENSOR
SW13 ACCELERATOR OPENING SENSOR
SW14 CAM ANGLE SENSOR (INTAKE-SIDE)
SW15 CAM ANGLE SENSOR (EXHAUST-SIDE)
SW16 FUEL PRESSURE SENSOR
DIRECT-INJECTION INJECTOR 67
SPARK PLUG 25
VVT 72
CVVL (INTAKE-SIDE) 74
VVL (EXHAUST-SIDE) 71
HIGH PRESSURE FUEL SUPPLY SYSTEM 62
THROTTLE VALVE 36
INTERCOOLER BYPASS VALVE 351
EGR VALVE 511
EGR COOLER BYPASS VALVE 531

FIG.15

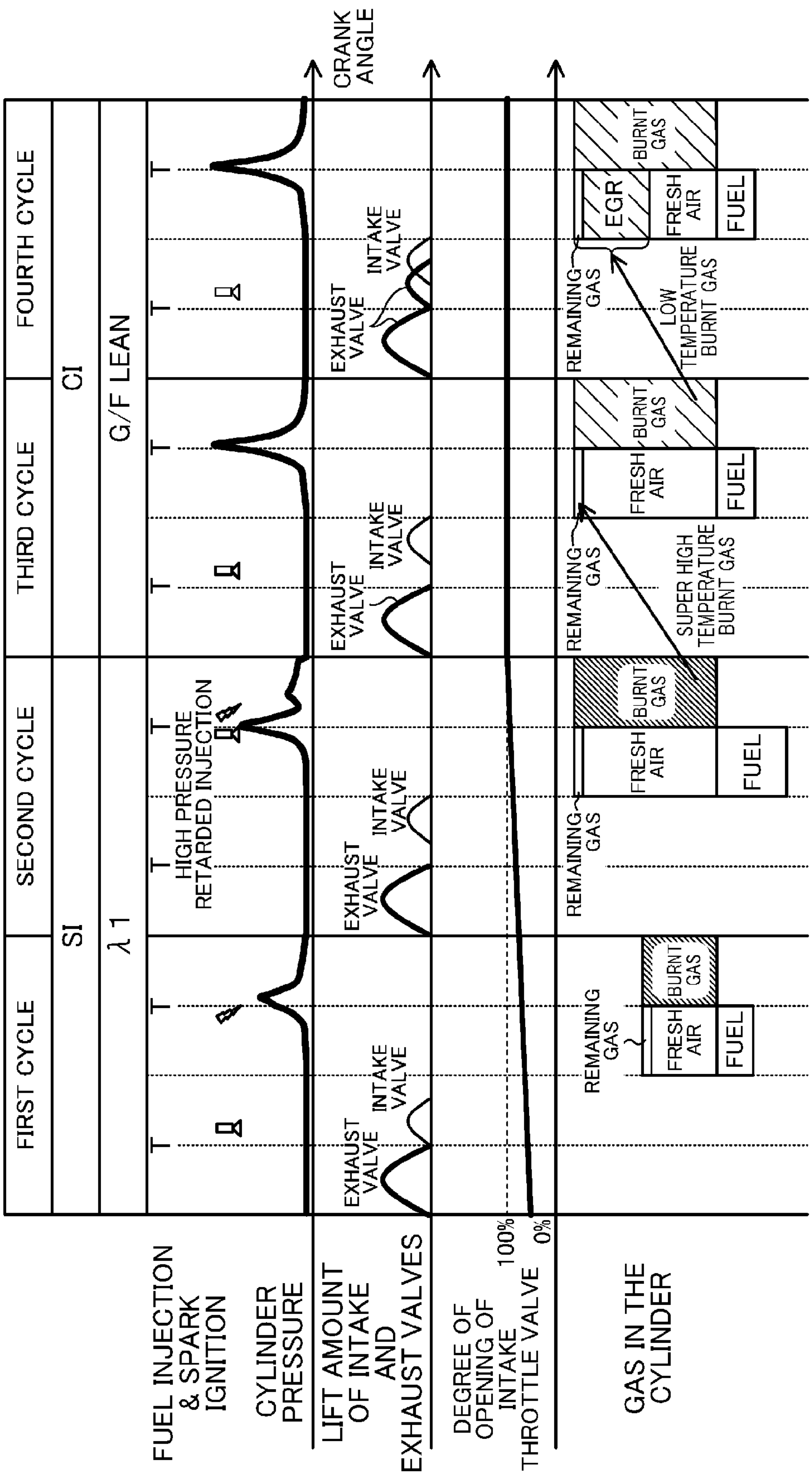

FIRST CYCLE
SECOND CYCLE
THIRD CYCLE
FOURTH CYCLE
SI
CI
λ1
G/F LEAN
CRANK ANGLE
HIGH PRESSURE RETARDED INJECTION
EXHAUST VALVE
INTAKE VALVE
100%
0%
REMAINING GAS
FRESH AIR
BURNT GAS
FUEL
EGR
SUPER HIGH TEMPERATURE BURNT GAS
LOW TEMPERATURE BURNT GAS
FUEL INJECTION & SPARK IGNITION
CYLINDER PRESSURE
LIFT AMOUNT OF INTAKE AND EXHAUST VALVES
DEGREE OF OPENING OF INTAKE THROTTLE VALVE
GAS IN THE CYLINDER

FIG.16

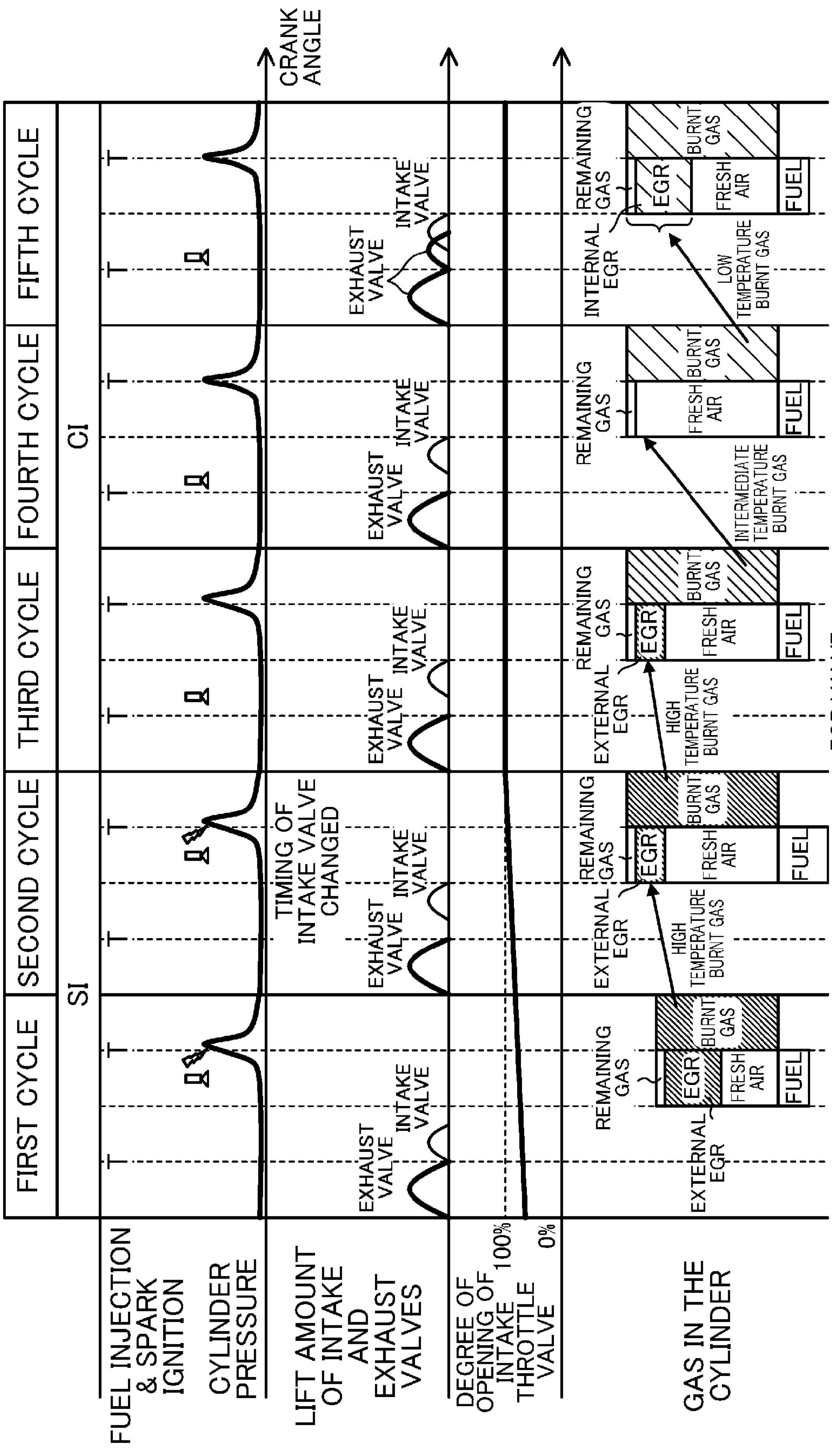
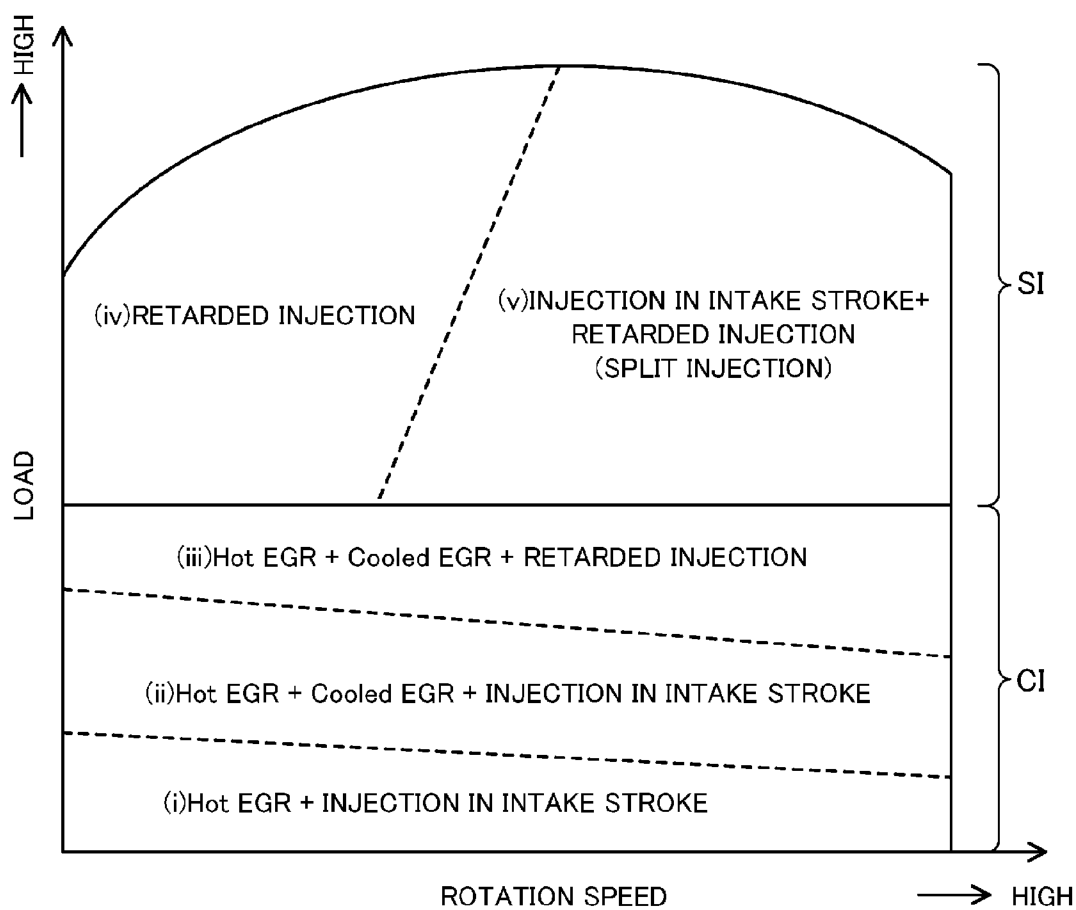

FIRST CYCLE
SECOND CYCLE
THIRD CYCLE
FOURTH CYCLE
FIFTH CYCLE
SI
CI
CRANK ANGLE
FUEL INJECTION & SPARK IGNITION
CYLINDER PRESSURE
LIFT AMOUNT OF INTAKE AND EXHAUST VALVES
DEGREE OF OPENING OF INTAKE THROTTLE VALVE
100%
0%
GAS IN THE CYLINDER
EXHAUST VALVE
INTAKE VALVE
TIMING OF INTAKE VALVE CHANGED
EXTERNAL EGR
INTERNAL EGR
REMAINING GAS
EGR
FRESH AIR
BURNT GAS
FUEL
HIGH TEMPERATURE BURNT GAS
INTERMEDIATE TEMPERATURE BURNT GAS
LOW TEMPERATURE BURNT GAS
EGR VALVE TURNED OFF

FIG.17


HIGH
LOAD
(iv)RETARDED INJECTION
(v)INJECTION IN INTAKE STROKE+
RETARDED INJECTION
(SPLIT INJECTION)
SI
(iii)Hot EGR + Cooled EGR + RETARDED INJECTION
(ii)Hot EGR + Cooled EGR + INJECTION IN INTAKE STROKE
CI
(i)Hot EGR + INJECTION IN INTAKE STROKE
ROTATION SPEED
HIGH

SPARK IGNITION ENGINE

TECHNICAL FIELD

The disclosed technology relates to a spark ignition engine.

BACKGROUND ART

As described in Patent Document 1, for example, a combustion pattern by compression ignition of an air-fuel mixture in a cylinder has been known as a technology for improving both of exhaust emission performance and thermal efficiency of an engine. However, the compression ignition combustion involves a drastic pressure increase with an increase in load of the engine, and leads to increase in combustion noise. Thus, as described in Patent Document 1, the compression ignition combustion engine generally employs, in place of the compression ignition, spark ignition by driving a spark plug in a high-load operation range.

Patent Document 2 describes an engine which performs the compression ignition combustion in a low-load, low-speed range like the engine of Patent Document 1. This engine keeps a high temperature burnt gas in the cylinder by adjusting a period in which an intake valve and an exhaust valve are open in a range of compression ignition combustion to increase the temperature in the cylinder and accelerate compression auto-ignition combustion. In a high-load, high-speed range in the range of compression ignition combustion, the engine advances the timing of opening the intake valve to blow the burnt gas in the cylinder back to an intake port, and then reintroduces the burnt gas into the cylinder together with fresh air. Thus, the fresh air reduces the temperature of the burnt gas, and a compression end temperature relatively decreases in the high-load, high-speed range, thereby reducing sudden pressure increase by the compression ignition combustion.

CITATION LIST

Patent Documents

[Patent Document 1] Japanese Unexamined Patent Publication No. 2007-154859

[Patent Document 2] Japanese Unexamined Patent Publication No. 2009-197740

SUMMARY OF THE INVENTION

Technical Problem

In the spark ignition combustion, the temperature of a combustion gas increases due to relatively low thermal efficiency. In the compression ignition combustion, as described in the patent documents mentioned above, the high temperature burnt gas is introduced into the cylinder to ensure ignitability. Accordingly, the temperature in the cylinder is relatively high immediately after the engine is switched from the spark ignition combustion to the compression ignition combustion, and the high temperature burnt gas generated by the spark ignition combustion is introduced into the cylinder to excessively increase the temperature in the cylinder. This may lead to preignition, i.e., the air-fuel mixture in the cylinder is compression-ignited in a compression stroke, for example, and a rate of pressure increase (dP/dθ) in the cylinder rapidly increases to make loud combustion noise.

The switching from the spark ignition combustion to the compression ignition combustion is performed not only when the load of the engine decreases, for example, but also when the load of the engine remains unchanged. The switching may also be performed, for example, when the temperature of the engine increases from a cold range to a warm range, or in other situations.

In view of the foregoing, the disclosed technology has been achieved to avoid an increase in combustion noise when the engine is switched from the spark ignition combustion to the compression ignition combustion.

Solution to the Problem

According to the disclosed technology, an EGR ratio, which is a ratio of an amount of an exhaust gas introduced into the cylinder, is temporarily reduced immediately after the engine is switched from the spark ignition combustion to the compression ignition combustion. This reduces an amount of a high temperature exhaust gas generated by the spark ignition combustion and introduced into the cylinder, and reduces the temperature in the cylinder immediately after the switching to the compression ignition combustion. Thus, the preignition of the air-fuel mixture is avoided, and the generation of the combustion noise is avoided.

Specifically, the disclosed technology relates to a spark ignition engine. The engine includes: an engine body having a cylinder; a spark plug disposed to face an inside of the cylinder, and configured to ignite an air-fuel mixture in the cylinder; an exhaust returning apparatus configured to introduce an exhaust gas into the cylinder; and a controller configured to operate the engine body by controlling at least the spark plug and the exhaust returning apparatus.

The controller performs switching between a compression ignition mode in which compression ignition combustion is performed by auto-ignition of the air-fuel mixture in the cylinder to operate the engine body, and a spark ignition mode in which spark ignition combustion is performed by driving the spark plug to ignite and combust the air-fuel mixture in the cylinder to operate the engine body, and the controller controls the exhaust returning apparatus at least in the compression ignition mode to introduce the exhaust gas into the cylinder so that an EGR ratio which is a ratio between an amount of the exhaust gas to a total amount of a gas in the cylinder is a predetermined value.

The controller reduces the EGR ratio to be lower than the EGR ratio set in the compression ignition mode to switch the engine body operated in the spark ignition mode to a transitional mode in which the compression ignition combustion is performed before switching to the compression ignition mode.

According to this configuration, the EGR ratio in the transitional mode employed in switching from the spark ignition mode to the compression ignition mode is lower than the EGR ratio set in the compression ignition mode. This reduces the amount of the high temperature exhaust gas generated in the spark ignition mode and introduced into the cylinder before the switching of the combustion mode. Thus, the temperature in the cylinder is relatively reduced in the transitional mode. "Reducing the EGR ratio to be lower than the EGR ratio set in the compression ignition mode" includes reducing the EGR ratio to zero so as not to introduce the high temperature exhaust gas into the cylinder.

The compression ignition combustion is performed in the transitional mode. Since the temperature in the cylinder is low in the transitional mode, preignition of the air-fuel mixture in the cylinder is avoided, and the air-fuel mixture is compression-ignited at appropriate timing. Thus, the compression ignition combustion with high thermal efficiency is performed with the temperature in the cylinder kept low to reduce the temperature of the combustion gas and the temperature of the exhaust gas in the transitional mode.

In the compression ignition mode after the transitional mode, the EGR ratio is relatively high. Accordingly, a relatively large amount of the exhaust gas is introduced into the cylinder. As described above, the temperature of the exhaust gas has been reduced in the transitional mode. Therefore, the temperature in the cylinder does not excessively increase in the compression ignition mode. As a result, the preignition is avoided, and the air-fuel mixture is compression-ignited at appropriate timing also in the compression ignition mode. This avoids the increase in combustion noise in switching the engine body from the spark ignition combustion to the compression ignition combustion.

The exhaust returning apparatus may include an internal EGR regulator for regulating an amount of the exhaust gas remaining in the cylinder in a period from an exhaust stroke to an intake stroke, and the controller may make the internal EGR regulator inactive in the spark ignition mode and the transitional mode, and may activate the internal EGR regulator in the compression ignition mode.

The internal EGR regulator is inactive in the spark ignition mode. This avoids the excessive increase in temperature in the cylinder, and advantageously avoids the preignition and knocking. The internal EGR regulator is activated in the compression ignition mode. This increases the temperature in the cylinder, and improves ignitability and stability of the compression ignition combustion.

The internal EGR regulator is inactive in the transitional mode. Thus, the internal EGR regulator remains inactive in switching from the spark ignition mode to the transitional mode. This allows smooth mode switching. The high temperature exhaust gas is not introduced into the cylinder in the transitional mode. This avoids the preignition in the transitional mode, and effectively avoids the generation of the combustion noise.

The spark ignition engine may further include a valve mechanism configured to control operation of an intake valve and an exhaust valve, wherein the valve mechanism may perform a first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke in the spark ignition mode, the valve mechanism, in addition to the first valve operation, may bring the exhaust valve in an open state at timing later than timing of closing the exhaust valve in the first valve operation, or may bring the intake valve in an open state at timing earlier than timing of opening the intake valve in the first valve operation in the compression ignition mode, and the valve mechanism may perform the first valve operation to operate the intake valve and the exhaust valve in the transitional mode.

In the compression ignition mode, in addition to the first valve operation in the spark ignition mode, the exhaust valve is in the open state at the timing later than the timing of closing the exhaust valve in the first valve operation. The exhaust valve is "in an open state" means that the exhaust valve which is opened in the exhaust stroke is kept open in the intake stroke (the exhaust valve is not closed), and that the exhaust valve which is opened in the exhaust stroke is once closed, and then reopened in the intake stroke. Keeping the exhaust valve which is opened in the exhaust valve in the open state in the intake stroke includes increasing a lift amount of the exhaust valve gradually decreased from a lift amount maximized in the exhaust stroke, and increasing a period in which the exhaust valve is open without increasing the maximized lift amount of the exhaust valve (e.g., maintaining the lift amount with respect to variation in crank angle, or reducing a ratio of reduction in lift amount). Thus, the exhaust valve can be opened in a way different from the first valve operation by using a cam shift mechanism for switching a plurality of cams having different cam characteristics. The cam shift mechanism is responsive to the switching of the valve, and is advantageous in controlling the switching of the combustion mode.

With the exhaust valve opened in the exhaust stroke and the intake stroke, part of the exhaust gas emitted from the cylinder to the exhaust side in the exhaust stroke is reintroduced into the cylinder when the exhaust valve is in the open state in the subsequent intake stroke. This increases the amount of the exhaust gas remaining in the cylinder in a period from the exhaust stroke to the intake stroke.

Likewise, in addition to the first valve operation in the spark ignition mode, the intake valve may be in an open state in the compression ignition mode at timing earlier than timing of opening the intake valve in the first valve operation. The intake valve is "in an open state" means the same as described above in connection with the exhaust valve.

In this configuration, the intake valve is in the open state in the exhaust stroke and the intake stroke. Thus, in the exhaust stroke, part of the exhaust gas is emitted from the cylinder to the intake side, and the exhaust gas is reintroduced into the cylinder when the intake valve is in the open state in the subsequent intake stroke. This also increases the amount of the exhaust gas remaining in the cylinder in the period from the exhaust stroke to the intake stroke.

In contrast to the valve operation in the compression ignition mode, the intake valve and the exhaust valve are operated by the first valve operation in the spark ignition mode in the transitional mode. Thus, the amount of the exhaust gas remaining in the cylinder in the period from the exhaust stroke to the intake stroke is ideally reduced to zero. Specifically, the EGR ratio in the transitional mode is reduced to be lower than the EGR ratio in the compression ignition mode.

In switching from the spark ignition mode to the compression ignition mode through the transitional mode, the valve mechanism may perform a first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke in the spark ignition mode, the valve mechanism may perform a second valve operation of opening the intake valve for a shorter period than a period for opening the intake valve in the first valve operation in the transitional mode, and the valve mechanism may perform, in addition to the second valve operation, a third valve operation of bringing the exhaust valve in an open state at timing later than timing of closing the exhaust valve in the second valve operation in the compression ignition mode.

In this configuration, the vale system changes the operation of opening the intake valve in switching from the spark ignition mode to the transitional mode. Specifically, in contrast to the first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke, the operation of opening the intake valve is changed to reduce the period in which the intake valve is open. The period in which the intake valve is open can be changed by a cam shift mechanism for switching a plurality of cams. The cam shift mechanism changes the period in which the intake valve is open with high responsiveness.

In switching from the transitional mode to the compression ignition mode, the valve mechanism changes the operation of opening the exhaust valve. Specifically, the exhaust valve opened in the exhaust stroke in the transitional mode is opened in the exhaust stroke and the intake stroke in the compression ignition mode. The switching of a lift characteristic of the valve can be achieved by the cam shift mechanism described above.

Thus, according to the above-described configuration, the valve mechanism changes the lift characteristic of the intake valve in switching the engine body from the spark ignition mode to the transitional mode, and then the valve mechanism changes the lift characteristic of the exhaust valve to switch the engine body from the transitional mode to the compression ignition mode. This enhances a response to the mode switching control, and is advantageous in smoothly switching from the spark ignition mode to the compression ignition mode.

In switching from the spark ignition mode to the compression ignition mode through the transitional mode, the valve mechanism may perform a first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke in the spark ignition mode, the valve mechanism may perform a second valve operation of opening the exhaust valve for a shorter period than a period for opening the exhaust valve in the first valve operation in the transitional mode, and the valve mechanism may perform, in addition to the second valve operation, a third valve operation of bringing the intake valve in an open state at timing earlier than timing of opening the intake valve in the second valve operation in the compression mode.

In this configuration, in a reverse manner of the above-described manner, the valve mechanism changes the operation of opening the exhaust valve in switching from the spark ignition mode to the transitional mode, and changes the operation of opening the intake valve in switching from the transitional mode to the compression ignition mode. This configuration also enhances the response to the mode switching control, and is advantageous in smoothly switching from the spark ignition mode to the compression ignition mode.

The controller may switch the engine body from the compression ignition mode to the spark ignition mode without switching the engine body to the transitional mode.

In switching from the compression ignition mode to the spark ignition mode, the combustion noise accompanying the preignition is not generated. Thus, the compression ignition mode can directly be switched to the spark ignition mode without switching the engine body to the transitional mode. Skipping the transitional mode allows quick switching from the compression ignition mode to the spark ignition mode. In the case where the compression ignition mode is employed when the load of the engine body is relatively low, and the spark ignition mode is employed when the load of the engine body is relatively high, the engine body is switched from the compression ignition mode to the spark ignition mode in accordance with a driver's request for acceleration. Thus, the quick switching from the compression ignition mode to the spark ignition mode enhances a response to acceleration of the vehicle.

The spark ignition engine may further include a fuel injection valve configured to directly inject a fuel in the cylinder, wherein the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

As described above, the occurrence of the preignition can be reduced by switching the engine body operated in the spark ignition mode to the transitional mode in which the EGR ratio is relatively reduced before switching to the compression ignition mode. However, the preignition may possibly occur in the transitional mode when the temperature in the cylinder is relatively high, e.g., when the temperature of an inner wall of the cylinder is relatively high, after the spark ignition mode is switched to the transitional mode.

The transitional mode may be continued to retard the timing of switching to the compression ignition mode until the temperature in the cylinder decreases. However, when the EGR ratio is kept relatively low for a long time, the temperature in the cylinder excessively decreases, and the compression ignition cannot stably be performed. Thus, although the engine body needs to be switched to the compression ignition mode for stable ignition, the preignition may possibly occur in the compression ignition mode due to the relatively high temperature of the inner wall of the cylinder.

Thus, in the transitional mode or the compression ignition mode, the fuel injection valve may inject the fuel at timing after the intermediate stage of the compression stroke in accordance with the temperature in the cylinder, e.g., when the temperate in the cylinder is higher than a predetermined temperature. The temperature in the cylinder decreases due to latent heat of vaporization of the fuel injected in the cylinder. Further, the timing of injecting the fuel is relatively retarded, and reactive time of the air-fuel mixture is reduced. Thus, the preignition can surely be avoided in the transitional mode or the compression ignition mode.

Advantages of the Invention

As described above, the spark ignition engine makes it possible to avoid increase in combustion noise in switching from the spark ignition mode to the compression ignition mode.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic view showing a configuration of a spark ignition direction injection engine.

FIG. **2** is a block diagram related to control of the spark ignition direction injection engine.

FIG. **3** is an enlarged cross-sectional view showing a combustion chamber.

FIG. **4** is a view illustrating an operation range of an engine.

FIG. **5**A shows an example of timing of fuel injection performed in an intake stroke in a CI mode, and a heat release rate in CI combustion accompanying the fuel injection.

FIG. **5**B shows an example of timing of high pressure retarded injection performed in the CI mode, and the heat release rate in the CI combustion accompanying the high pressure retarded injection.

FIG. **5**C shows an example of timing of the high pressure retarded injection performed in a SI mode and timing of ignition, and the heat release rate in SI combustion accompanying the high pressure retarded injection.

FIG. **5**D shows an example of timing of split injection including the injection in the intake stroke and the high pressure retarded injection and timing of ignition, and the heat release rate in the SI combustion accompanying the split injection.

FIG. 6 is a view showing comparison between SI combustion by the high pressure retarded injection and conventional SI combustion.

FIG. 7 is a timing chart illustrating transitional control in switching from an SI mode in which an external EGR gas is not introduced to a CI mode.

FIG. 8 is a timing chart illustrating transitional control in switching from the SI mode in which an external EGR gas is introduced to the CI mode.

FIG. 9 is a flow chart illustrating transitional control in switching from the SI mode to the CI mode.

FIG. 10 is a view illustrating lift amounts of an intake valve and an exhaust valve associated with the exhaust valve opened twice.

FIG. 11 is a view illustrating the lift amounts of the intake valve and the exhaust valve associated with the exhaust valve opened twice and different from the lift amounts of FIG. 10.

FIG. 12 shows the lift amounts of the intake valve and the exhaust valve associated with the exhaust valve opened twice and different from the lift amounts of FIGS. 10 and 11.

FIG. 13 shows the lift amounts of the intake valve and the exhaust valve associated with the intake valve opened twice.

FIG. 14 is a block diagram illustrating a spark ignition direct injection engine having a configuration different from the configuration shown in FIG. 2.

FIG. 15 is a timing chart illustrating transitional control in switching from the SI mode in which the external EGR gas is not introduced to the CI mode different from the transitional control of FIG. 7.

FIG. 16 is a timing chart illustrating transitional control in switching from the SI mode in which the external EGR gas is introduced to the CI mode different from the transitional control of FIG. 8.

FIG. 17 shows an operation range of the engine different from the operation range of FIG. 4.

DESCRIPTION OF EMBODIMENTS

An embodiment of a spark ignition engine will be described with reference to the drawings. The following preferred embodiment is provided only for the illustration purpose.

(General Configuration of Engine)

FIGS. 1 and 2 show a schematic configuration of an engine (an engine body) 1. The engine 1 is a spark ignition gasoline engine mounted on a vehicle, and supplied with a fuel containing at least gasoline. The engine 1 includes a cylinder block 11, a cylinder head 12 arranged on the cylinder block 11, and an oil pan 13 arranged below the cylinder block 11. The cylinder block 11 is provided with a plurality of cylinders 18. FIG. 1 shows a single cylinder only, but the engine 1 includes four cylinders arranged in series, for example.

A piston 14 is reciprocatably fitted in each of the cylinders 18. The piston 14 is coupled to a crankshaft 15 through a connecting rod 142. As shown in an enlargement in FIG. 3, a cavity 141 like a re-entrant cavity of a diesel engine is formed in a top face of the piston 14. The cavity 141 faces an injector 67 described later when the piston 14 is located near a compression top dead center. The cylinder head 12, the cylinder 18, and the piston 14 having the cavity 141 define a combustion chamber 19. The shape of the combustion chamber 19 is not limited to the illustrated shape. The shapes of the cavity 141, the top face of the piston 14, a ceiling of the combustion chamber 19, etc. can be changed as appropriate.

The engine 1 has a relatively high geometric compression ratio of 15 or higher for improved theoretical thermal efficiency, stabilized compression ignition combustion described later, etc. The geometric combustion ratio may be set as appropriate within a range from 15 or higher to 20 or lower. For example, the geometric combustion ratio of the engine 1 is 18.

The cylinder head 12 has an intake port 16 and an exhaust port 17 formed for each of the cylinders 18. An intake valve 21 and an exhaust valve 22 are arranged in the intake port 16 and the exhaust port 17, respectively, to open/close openings near the combustion chamber 19.

A valve mechanism for driving the exhaust valve 22 includes, for example, a hydraulic variable mechanism (hereinafter referred to as a variable valve lift (VVL) mechanism, see FIG. 2) 71. The VVL 71 switches an operation mode of the exhaust valve 22 between a normal mode and a special mode. Although its configuration is not illustrated in detail, the VVL 71 on the exhaust side includes two different types of cams having different cam profiles, i.e., first and second cams, and a lost motion mechanism for selectively transmitting an operation state of one of the first cam or the second cam to the exhaust valve. The two different types of cams include the first cam having a single cam lobe and the second cam having two cam lobes. When the VVL 71 transmits the operation state of the first cam to the exhaust valve 22, the exhaust valve 22 is operated in the normal mode to be opened only once in an exhaust stroke. When the VVL 71 transmits the operation state of the second cam to the exhaust valve 22, the exhaust valve 22 is opened in the exhaust stroke and in an intake stroke. Specifically, the exhaust valve 22 is operated in the special mode to be opened twice (see FIG. 10, etc.). The exhaust-side VVL 71 switches the operation mode between the normal mode and the special mode in accordance with the operation state of the engine 1. Specifically, the special mode is used in controlling internal EGR. In the following description, operating the exhaust-side VVL 71 in the normal mode so as not to open the exhaust valve 22 twice may be referred to as "turn the VVL 71 off," and operating the exhaust-side VVL 71 in the special mode to open the exhaust valve 22 twice may be referred to as "turn the VVL 71 on." For the switching between the normal mode and the special mode, an electromagnetic valve mechanism for driving the exhaust valve 22 using an electromagnetic actuator may be employed. The internal EGR is performed not only by opening the exhaust valve 22 twice. For example, the internal EGR may be controlled by opening the intake valve 21 twice. The internal EGR may be controlled by leaving burnt gas in the cylinder 18 by setting a negative overlap period in which both of the intake valve 21 and the exhaust valve 22 are closed in the exhaust and intake strokes.

A valve mechanism for driving the intake valve 21 also includes a VVL 73 like the exhaust valve mechanism including the VVL 71. The intake-side VVL 73 is different from the exhaust-side VVL 71. The intake-side VVL 73 includes two different types of cams having different cam profiles, i.e., a large lift cam which relatively increases a lift amount of the intake valve 21, and a small lift cam which relatively decreases the lift amount of the intake valve 21, and a lost motion mechanism for selectively transmitting an operation state of one of the large lift cam or the small lift cam to the intake valve 21. As shown in FIG. 10, etc., when the intake-side VVL 73 transmits the operation state of the large lift cam to the intake valve 21, the intake valve 21 is opened with a relatively large lift amount for a long opening time. When the intake-side VVL 73 transmits the operation state of the small lift cam to the intake valve **21**, the intake valve **21** is opened with a relatively small lift amount for a short opening time. The intake-side valve mechanism further includes a phase variable mechanism (hereinafter referred to as a variable valve timing (VVT) mechanism) **72** as shown in FIG. **2**. The VVT **72** changes a rotation phase of an intake camshaft relative to the crankshaft **15**. The VVT **72** may have a known hydraulic, electromagnetic, or mechanical configuration as appropriate, and details thereof are not shown in the drawings. The VVT **72** and the VVL **73** can change timings of opening and closing the intake valve **21**, and the lift amount of the intake valve **21**.

The cylinder head **12** is further provided with injectors **67** for directly injecting the fuel in the cylinders **18**, respectively. As shown in the enlargement in FIG. **3**, each of the injectors **67** is arranged with its nozzle facing the inside of the combustion chamber **19** from a center of a ceiling of the combustion chamber **19**. The injector **67** directly injects the fuel in an amount in accordance with the operation state of the engine at injection timing determined in accordance with the operation state of the engine **1**. In the illustrated example, the injector **67** is a multi-hole injector having a plurality of nozzles. Thus, the injector **67** injects the fuel such that the fuel is radially sprayed from the center of the combustion chamber **19**. As indicated by arrows in FIG. **3**, the fuel radially sprayed from the center of the combustion chamber **19** when the piston **14** is located near the compression top dead center flows along a wall surface of the cavity **141** formed in the top face of the piston. In other words, the cavity **141** is shaped to contain the fuel sprayed when the piston **14** is located near the compression top dead center. A combination of the multi-hole injector **67** and the cavity **141** is advantageous in shortening a period for forming the air-fuel mixture after the spraying of the fuel, and shortening a combustion period. The injector **67** is not limited to the multi-hole injector, and an injector having an outward lifting valve may be used.

A fuel tank not shown in the drawings and the injector **67** are connected by a fuel supply passage. A fuel supply system **62** is provided in the fuel supply passage. The fuel supply system **62** includes a fuel pump **63** and a common rail **64**, and can supply the fuel to the injector **67** at a relatively high fuel pressure. The fuel pump **63** pumps the fuel from the fuel tank to the common rail **64**. The common rail **64** stores the pumped fuel at a relatively high fuel pressure. The injector **67** injects the fuel stored in the common rail **64** from the nozzle. Although not shown, the fuel pump **63** is a plunger pump, and is driven by the engine **1**. The fuel supply system **62** including the engine-driven pump can supply the fuel at a high fuel pressure of 30 MPa or higher to the injector **67**. The fuel pressure may be set to about 120 MPa at the maximum. The pressure of the fuel supplied to the injector **67** is varied in accordance with the operation state of the engine **1**. The fuel supply system **62** is not limited to this configuration.

As shown in FIG. **3**, the cylinder head **12** further includes a spark plug **25** for igniting the air-fuel mixture in the combustion chamber **19**. In this example, the spark plug **25** penetrates the cylinder head **12** to extend obliquely downward from the exhaust side of the engine **1**. As shown in FIG. **3**, a tip of the spark plug **25** is disposed in the cavity **141** of the piston **14** located at the compression top dead center.

An intake air passage **30** is connected to one side surface of the engine **1** as shown in FIG. **1**. The intake air passage **30** communicates with the intake port **16** of each of the cylinders **18**. An exhaust air passage **40** is connected to the other side surface of the engine **1**. The exhaust air passage **40** emits the burnt gas (exhaust gas) from the combustion chamber **19** of each of the cylinders **18**.

An air cleaner **31** for filtering intake air is disposed at an upstream end of the intake air passage **30**. A surge tank **33** is disposed near a downstream end of the intake air passage **30**. Part of the intake air passage **30** downstream of the surge tank **33** is divided into independent passages corresponding to the cylinders **18**. Downstream ends of the independent passages are connected to the intake ports **16** of the cylinders **18**.

An intercooler/warmer **34** and a throttle valve **36** are disposed in the intake air passage **30** between the air cleaner **31** and the surge tank **33**. The intercooler/warmer **34** is a water-cooling cooler, for example. The intercooler/warmer **34** cools or heats the air. The throttle valve **36** regulates an amount of the intake air supplied to each of the cylinders **18**. An intercooler bypass passage **35** for bypassing the intercooler/warmer **34** is connected to the intake air passage **30**. The intercooler bypass passage **35** is provided with an intercooler bypass valve **351**. The intercooler bypass valve **351** regulates a flow rate of the air passing through the intercooler bypass passage **35**. Adjusting the degree of opening of the intercooler bypass valve **351** adjusts a ratio between the flow rate of the air passing through the intercooler bypass passage **35** and the flow rate of the air passing through the intercooler/warmer **34**. This makes it possible to regulate the temperature of fresh air introduced into the cylinders **18**.

An upstream part of the exhaust air passage **40** is constituted of an exhaust manifold. The exhaust manifold includes independent passages for the cylinders **18** connected to outer ends of the exhaust ports **17**, and an assembly part in which the independent passages are gathered. As an exhaust emission purifier for purifying hazardous components in the exhaust gas, a direct catalyst **41** and an underfoot catalyst **42** are connected to part of the exhaust air passage **40** downstream of the exhaust manifold. Each of the direct catalyst **41** and the underfoot catalyst **42** includes a tubular case, and a three-way catalyst disposed in a passage in the case, for example.

Part of the intake air passage **30** between the surge tank **33** and the throttle valve **36** is connected to part of the exhaust air passage **40** upstream of the direct catalyst **41** through an EGR passage **50**. The EGR passage **50** returns part of the exhaust gas to the intake air passage **30**. The EGR passage **50** includes a main passage **51** in which an EGR cooler **52** is disposed, and an EGR cooler bypass passage **53** for bypassing the EGR cooler **52**. The EGR cooler **52** cools the exhaust gas with engine cooling water. The main passage **51** is provided with an EGR valve **511**. The EGR valve **511** regulates an amount of the exhaust gas returned to the intake air passage **30**. The EGR cooler bypass passage **53** is provided with an EGR cooler bypass valve **531**. The EGR cooler bypass valve **531** regulates a flow rate of the exhaust gas circulating through the EGR cooler bypass passage **53**.

The engine **1** is controlled by a powertrain control module (hereinafter referred to as PCM) **10**. The PCM **10** is a microprocessor including a CPU, a memory, a counter timer group, an interface, and paths connecting these units. The PCM **10** constitutes a controller.

As shown in FIGS. **1** and **2**, the PCM **10** receives detection signals from various sensors SW**1**-SW**16**. The various sensors include the following sensors. Specifically, an air flow sensor SW**1** and an intake air temperature sensor SW**2** are arranged downstream of the air cleaner **31** to detect the flow rate and the temperature of the fresh air, respectively. A second intake air temperature sensor SW**3** is arranged downstream of the intercooler/warmer **34** to detect the temperature of the fresh air passed through the intercooler/warmer **34**. An EGR gas temperature sensor SW**4** is arranged near a junction between the EGR passage **50** and the intake air passage **30** to detect the temperature of external EGR gas. An intake port temperature sensor SW**5** is attached to the intake port **16** to detect the temperature of the intake air immediately before flowing into the cylinder **18**. A cylinder pressure sensor SW**6** is attached to the cylinder head **12** to detect the pressure in the cylinder **18**. An exhaust gas temperature sensor SW**7** and an exhaust gas pressure sensor SW**8** are arranged near a junction between the exhaust air passage **40** and the EGR passage **50** to detect the temperature and the pressure of the exhaust gas, respectively. A linear O_2 sensor SW**9** is arranged upstream of the direct catalyst **41** to detect a concentration of oxygen in the exhaust gas. A lambda O_2 sensor SW**10** is arranged between the direct catalyst **41** and the underfoot catalyst **42** to detect the concentration of oxygen in the exhaust gas. A water temperature sensor SW**11** detects the temperature of the engine cooling water. A crank angle sensor SW**12** detects a rotation angle of the crankshaft **15**. An accelerator opening sensor SW**13** detects a degree of opening of an accelerator corresponding to a position of an accelerator pedal (not shown) of the vehicle. A cam sensor SW**14** is provided on the exhaust side, and a cam sensor SW**15** is provided on the intake side. A fuel pressure sensor SW**16** is attached to the common rail **64** of the fuel supply system **62** to detect the pressure of the fuel supplied to the injector **67**.

The PCM **10** performs various types of calculation based on these detection signals to determine the conditions of the engine **1** and the vehicle. In accordance with the results of the determination, the PCM **10** outputs control signals to actuators of the injector **67**, the spark plug **25**, the VVT **72**, the intake-side VVL **73**, the exhaust-side VVL **71**, the fuel supply system **62**, and the valves (the throttle valve **36**, the intercooler bypass valve **351**, the EGR valve **511**, and the EGR cooler bypass valve **531**). The PCM **10** operates the engine **1** in this way.

(Operation Range of Engine)

FIG. **4** shows an example operation range of the engine **1** in a warm state. For enhanced fuel efficiency and improved exhaust emission performance, the engine **1** does not perform ignition with the spark plug **25**, but performs compression ignition by compression auto-ignition in a low load range where the engine has a relatively low load. However, with an increase in load of the engine **1**, the compression ignition combustion proceeds too rapidly to cause problems, such as combustion noise. Thus, the engine **1** does not perform the compression ignition combustion, and performs the spark ignition combustion using the spark plug **25** in a high load range where the engine has a relatively high load. The PCM **10** switches an ignition mode between a compression ignition (CI) mode in which the compression ignition combustion is performed, and a spark ignition (SI) mode in which the spark ignition combustion is performed in accordance with the operation state of the engine **1**, particularly in accordance with the level of the load of the engine **1**. A boundary between the modes is not limited to the illustrated boundary. The PCM **10** may perform the mode switching in accordance with various situations described later in addition to the level of the load.

As shown in FIG. **4**, an operation range in the CI mode is divided into two ranges in accordance with the level of the load of the engine. In a range (I) where the load is low or medium in the operation range in the CI mode, the PCM **10** introduces hot EGR gas at relatively high temperature into the cylinder **18** to enhance ignitability and stability of the compression ignition combustion. Specifically, the PCM **10** turns the exhaust-side VVL **71** on to open the exhaust valve **22** twice in the exhaust stroke and the intake stroke. Introducing the hot EGR gas increases the compression end temperature in the cylinder **18**, and enhances the ignitability and the stability of the compression ignition combustion in the range (I). In the range (I), as shown in FIG. **5**A, the injector **67** injects the fuel in the cylinder **18** at least in a period from the intake stroke to an intermediate stage of a compression stroke. Thus, a homogeneous air-fuel mixture is formed in the cylinder **18**. In the range (I), an air-fuel ratio (A/F) of the air-fuel mixture is basically a theoretical air-fuel ratio (A/F=14.7±0.5, an excess air ratio: $\lambda \approx 1$). As indicated by a dot-and-dash line in FIG. **4**, the air-fuel mixture shows the A/F lean of the theoretical air-fuel ratio in part of the range (I) where the load is relatively low and an engine speed is relatively low.

In the range (I), the PCM **10** makes the spark plug **25** inactive. As shown in FIG. **5**A, compression auto-ignition of the air-fuel mixture in the chamber **19** occurs near the compression top dead center.

In the operation range in the CI mode, the A/F of the air-fuel mixture is the theoretical air-fuel ratio ($\lambda \approx 1$) in a range (II) where the load is higher than the load in the range (I). Thus, a three-way catalyst can be used. As described later, the A/F of the air-fuel mixture is the theoretical air-fuel ratio also in an operation range in the SI mode. Thus, in the range (II), the A/F of the air-fuel mixture being the theoretical air-fuel ratio contributes to simplification of the control in switching the ignition mode between the SI mode and the CI mode, and expansion of the operation range in the CI mode toward the high load range.

The temperature in the cylinder spontaneously increases with the increase in load of the engine. Thus, the PCM **10** reduces the amount of the hot EGR gas to avoid preignition in the range (II). The amount of the hot EGR gas is adjusted by adjusting an amount of internal EGR gas introduced into the cylinder **18**. The amount of the hot EGR gas can be adjusted by adjusting an amount of the external EGR gas bypassed the EGR cooler **52**.

In the range (II), the PCM **10** introduces cooled EGR gas at relatively low temperature into the cylinder **18**. Thus, the high temperature hot EGR gas and the low temperature cooled EGR gas are introduced into the cylinder **18** in an appropriate ratio. This optimizes the compression end temperature in the cylinder **18**, avoids rapid combustion while ensuring the ignitability of the compression ignition, and stabilizes the compression ignition combustion.

Thus, the PCM **10** is supposed to decrease the temperature in the cylinder **18** in the range (II) including the boundary between the CI mode and the SI mode. Nevertheless, the compression end temperature in the cylinder **18** may possibly increase. Accordingly, when the fuel is injected in the cylinder **18** in a period from the intake stroke to the intermediate stage of the compression stroke in the range (II), abnormal combustion, such as preignition, etc., may possibly occur. When a large amount of the low temperature cooled EGR gas is introduced to decrease the compression end temperature in the cylinder to avoid the preignition, the ignitability of the compression ignition may deteriorate in turn. Specifically, in the range (II), the compression ignition combustion cannot stably be performed simply by controlling the temperature in the cylinder **18**. Thus, in the range (II), the PCM **10** performs creative fuel injection in addition to the control of the temperature in the cylinder **18**, thereby avoiding the abnormal combustion such as preignition, etc., and stabilizing the compression ignition combustion. Specifically, in the creative fuel injection, the fuel at a significantly increased pressure as compared with a conventionally employed pressure is injected in the cylinder **18** at least in a period from a terminal stage of the compression stroke to an initial stage of an expansion stroke (hereinafter the period is referred to as a retarded period) as shown in FIG. **5**B. This distinctive fuel injection is hereinafter referred to as "high pressure retarded injection" or simply "retarded injection." The high pressure retarded injection makes it possible to avoid the abnormal combustion in the range (II), and to stabilize the compression ignition combustion. Details of the high pressure retarded injection will be described later.

While the operation range in the CI mode is divided into the two ranges in accordance with the level of the load of the engine, the operation range in the SI mode is divided into two ranges, i.e., a range (III) and a range (IV), roughly in accordance with the level of an engine speed. In the illustrated example, suppose that the operation range of the engine **1** is divided into a low speed range and a high speed range, the range (III) corresponds to a low speed range and a low load part in the high speed range, and the range (IV) corresponds to a high load part in the high speed range. A boundary between the range (III) and the range (IV) is not limited to the illustrated boundary.

In each of the range (III) and the range (IV), the A/F of the air-fuel mixture is the theoretical air-fuel ratio ($\lambda \approx 1$) like the A/F in the range (II). Thus, the A/F of the air-fuel mixture is constant at the theoretical air-fuel ratio ($\lambda \approx 1$) across the boundary between the CI mode and the SI mode. In the SI mode (i.e., in the range (III) and the range (IV)), the PCM **10** basically fully opens the throttle valve **36**, and adjusts the degree of opening of the EGR valve **511**. Thus, the PCM **10** adjusts a ratio of an amount of fresh air introduced into the cylinder **18** and an amount of the external EGR gas. Also in the operation range in the SI mode, the PCM **10** may reduce the degree of opening of the throttle valve **36** in a range where the load is relatively low. Adjusting the amount of the fresh air by adjusting the ratio of the gas introduced into the cylinder **18** reduces pump losses. Further, introducing a large amount of the EGR gas into the cylinder **18** keeps the combustion temperature in the spark ignition combustion low, thereby reducing cooling losses. In the operation range in the SI mode, the PCM **10** introduces the external EGR gas which is cooled mainly by the EGR cooler **52** into the cylinder **18**. This is advantageous in avoiding the abnormal combustion, and in reducing generation of raw NOx. In a full open load range, the PCM **10** closes the EGR valve **511** to reduce the amount of the external EGR gas to zero.

In the SI mode range, the PCM **10** stops introducing the EGR gas, and controls the degree of opening of the throttle valve **36** in accordance with the amount of the fuel injected by the injector **67** to adjust the amount of the fresh air introduced into the cylinder **18** such that the air-fuel mixture has the theoretical air-fuel ratio ($\lambda \approx 1$).

The geometric compression ratio of the engine **1** is set to 15 or higher (e.g., 18) as described above. A high compression ratio increases the compression end temperature and the compression end pressure. Thus, particularly in the low load range in the CI mode range (e.g., the range (I)), the engine **1** with the high compression ratio advantageously stabilizes the compression ignition combustion. On the other hand, the engine **1** with the high compression ratio may easily cause the abnormal combustion, such as preignition, knocking, etc., in the SI mode range which is a high load range.

Thus, the PCM **10** performs the above-described high pressure retarded injection in the range (III) and the range (IV) in the SI mode to avoid the abnormal combustion. More specifically, in the range (III), the injector **67** injects the fuel in the cylinder **18** at a high fuel pressure of 30 MPa or higher in the retarded period from the terminal stage of compression stroke to the initial stage of the expansion stroke as shown in FIG. **5**C. The injector **67** performs only the high pressure retarded injection in the range (III). In the range (IV), as shown in FIG. **5**D, the injector **67** injects part of the fuel in the cylinder **18** in a period of the intake stroke in which the intake valve **21** is open, and injects the remaining fuel in the cylinder **18** in the retarded period. That is, the injector **67** performs split fuel injection in the range (IV). The "period of the intake stroke in which the intake valve **21** is open" is a period defined not based on the position of the piston, but based on the opening/closing state of the intake valve **21**. Thus, timing when the "intake stroke" finishes may differ from timing when the piston reaches an intake bottom dead center in accordance with timing of closing the intake valve **21**, which is changed by the VVL **73** or the VVT **72**.

Referring to FIG. **6**, the high pressure retarded injection in the SI mode will be described below. FIG. **6** is a graph showing a comparison between SI combustion by the above-described high pressure retarded injection (indicated by a solid line) and conventional SI combustion by injecting the fuel in the period of the intake stroke (indicated by a broken line) in terms of a heat release rate (an upper portion of the graph) and reactivity of an unburnt air-fuel mixture (a lower portion of the graph). In FIG. **6**, a horizontal axis represents a crank angle. The comparison is based on the assumption that the operation state of the engine **1** is in a high-load, low-speed range (i.e., the range (III)), and the same amount of fuel is injected in the SI combustion by the high pressure retarded injection and in the conventional SI combustion.

In the conventional SI combustion, the injector **67** injects a predetermined amount of the fuel in the cylinder **18** in the period of the intake stroke (indicated by a broken line in the upper portion of the graph). In the cylinder **18**, a relatively homogeneous air-fuel mixture is formed before the piston **14** reaches the compression top dead center after the fuel injection. In the illustrated example, the spark plug **25** ignites the air-fuel mixture in the cylinder **18** at predetermined timing indicated by an open circle after the compression top dead center to start the combustion. After the start of the combustion, the heat release rate reaches its peak, and then the combustion finishes as indicated by the broken line in the upper portion of the graph of FIG. **6**. A period from the start of the fuel injection to the end of the combustion corresponds to reactive time of the unburnt air-fuel mixture (hereinafter may simply be referred to as reactive time). As indicated by a broken line in the lower portion of the graph of FIG. **6**, the reaction of the unburnt air-fuel mixture gradually proceeds for the reactive time. A dotted line in FIG. **6** indicates an ignition threshold of the reactivity of the unburnt air-fuel mixture at which the unburnt air-fuel mixture is ignited. Since the conventional SI combustion is performed in the low speed range, the reactive time is very long, for which the reaction of the unburnt air-fuel mixture continues. Thus, in the conventional SI combustion, the reactivity of the unburnt air-fuel mixture exceeds the ignition threshold before and after the ignition, thereby causing the abnormal combustion, such as preignition, knocking, etc.

On the other hand, the high pressure retarded injection is intended to avoid the abnormal combustion by reducing the reactive time. Specifically, as shown in FIG. **6**, the reactive time is a sum of a period for which the injector **67** injects the fuel (i.e., (1) an injection period), a period from when the injection finishes to when a burnable air-fuel mixture is formed around the spark plug **25** (i.e., (2) an air-fuel mixture formation period), and a period until the combustion started by the ignition finishes (i.e., (3) a combustion period), that is, (1)+(2)+(3). The high pressure retarded injection reduces each of the injection period, the air-fuel mixture formation period, and the combustion period to reduce the reactive time. This will be described in detail below.

A high fuel pressure relatively increases the amount of the fuel injected by the injector **67** per unit time. Thus, when the amount of the injected fuel is kept constant, the fuel pressure and the fuel injection period roughly establish a relationship in which the injection period increases with decrease in fuel pressure, i.e., the injection period decreases with increase in fuel pressure. Therefore, the high pressure retarded injection in which the fuel pressure is set significantly higher than the conventional fuel pressure reduces the injection period.

The high fuel pressure advantageously atomizes the fuel sprayed in the cylinder **18**, and increases a flight distance of the sprayed fuel. Thus, the fuel pressure and fuel evaporation time roughly establish a relationship in which the fuel evaporation time increases with the decrease in fuel pressure, i.e., the fuel evaporation time decreases with the increase in fuel pressure. Further, the fuel pressure and time taken until the sprayed fuel reaches the vicinity of the spark plug **25** roughly establish a relationship in which the time increases with the decrease in fuel pressure, i.e., the time decreases with the increase in fuel pressure. Since the air-fuel mixture formation period is a sum of the fuel evaporation time and the time taken until the sprayed fuel reaches the vicinity of the spark plug **25**, the air-fuel mixture formation period decreases with the increase in fuel pressure. Therefore, the high pressure retarded injection in which the fuel pressure is set significantly higher than the conventional fuel pressure reduces the fuel evaporation time and the time taken until the sprayed fuel reaches the vicinity of the spark plug **25**, thereby reducing the air-fuel mixture formation period. By contrast, as indicated by open circles in FIG. **6**, the conventional fuel injection at a low fuel pressure in the intake stroke significantly increases the air-fuel mixture formation period. In the SI mode, the above-described combination of the multi-hole injector **67** and the cavity **141** reduces the time taken until the sprayed fuel reaches the vicinity of the spark plug **25** after the fuel injection, thereby advantageously reducing the air-fuel mixture formation period.

Reducing the injection period and the air-fuel mixture formation period makes it possible to relatively retard the timing of the fuel injection, more precisely, the timing of the start of the injection. In the high pressure retarded injection, the fuel is injected in the cylinder **18** in the retarded period from the terminal stage of the compression stroke to the initial stage of the expansion stroke as shown in the upper portion of the graph of FIG. **6**. Since the fuel is injected in the cylinder **18** at the high fuel pressure, turbulence in the cylinder increases, and turbulence energy increases in the cylinder **18**. The high turbulence energy advantageously reduces the combustion period in combination with the relatively retarded timing of the fuel injection.

Specifically, when the fuel is injected in the retarded period, the fuel pressure and the turbulence energy in the combustion period roughly establish a relationship in which the turbulence energy decreases with the decrease in fuel pressure, i.e., the turbulence energy increases with the increase in fuel pressure. Suppose that the fuel is injected in the cylinder **18** at high fuel pressure in the period of the intake stroke, the turbulence in the cylinder **18** is reduced because it takes a long time until the air-fuel mixture is

ignited after the start of the injection, and the inside of the cylinder **18** is compressed in the compression stroke after the intake stroke. As a result, when the fuel is injected in the period of the intake stroke, the turbulence energy in the combustion period relatively decreases, irrespective of the level of the fuel pressure.

The turbulence energy in the combustion period and the combustion period roughly establish a relationship in which the combustion period increases with decrease in turbulence energy, i.e., the combustion period decreases with increase in turbulence energy. Accordingly, the fuel pressure and the combustion period establish a relationship in which the combustion period increases with the decrease in fuel pressure, i.e., the combustion period decreases with the increase in fuel pressure. Specifically, the high pressure retarded injection reduces the combustion period. By contrast, the conventional fuel injection at low fuel pressure in the intake stroke increases the combustion period. The multi-hole injector **67** advantageously increases the turbulence energy in the cylinder **18**, and advantageously reduces the combustion period. Further, combining the multi-hole injector **67** and the cavity **141** to contain the sprayed fuel in the cavity **141** is also advantageous in reducing the combustion period.

As described above, the high pressure retarded injection reduces each of the injection period, the air-fuel mixture formation period, and the combustion period. As a result, as shown in FIG. **6**, the reactive time of the unburnt air-fuel mixture from the start of the fuel injection SOI to the end of the combustion θend is significantly reduced as compared with the case where the conventional fuel injection is performed in the period of the intake stroke. In the conventional fuel injection performed at the low fuel pressure in the intake stroke, the reactivity of the unburnt air-fuel mixture at the end of the combustion exceeds the ignition threshold indicated by an open circle in the upper portion of the graph of FIG. **6**, thereby causing the abnormal combustion. On the other hand, the high pressure retarded injection reduces the reactive time, thereby slowing the reaction of the unburnt air-fuel mixture at the end of the combustion, and avoiding the abnormal combustion as indicated by a solid circle. In the injection indicated by the open and solid circles in the upper portion of the graph of FIG. **6**, the air-fuel mixture is ignited at the same time.

The fuel pressure set to 30 MPa or higher, for example, can effectively reduce the combustion period. The fuel pressure of 30 MPa or higher can effectively reduce both of the injection period and the air-fuel mixture formation period. The fuel pressure is preferably set as appropriate in accordance with the properties of the fuel used containing at least gasoline. An upper limit of the fuel pressure may be 120 MPa, for example.

The high pressure retarded injection avoids the occurrence of the abnormal combustion in the SI mode by devising the fuel injection in the cylinder **18**. As contrasted with this fuel injection, retarding the timing of the ignition for avoiding the abnormal combustion has been known. The retardation of the ignition timing slows the reaction of the unburnt air-fuel mixture by reducing the increase in temperature and pressure of the unburnt air-fuel mixture. However, the retardation of the ignition timing leads to decrease in thermal efficiency and torque. In contrast, the high pressure retarded injection can advance the ignition timing by avoiding the abnormal combustion by the devised fuel injection, thereby improving the thermal efficiency and the torque. Specifically, the high pressure retarded injection not only avoids the abnormal combustion, but also advances the ignition timing to advantageously improve the fuel efficiency.

As described above, the high pressure retarded injection in the SI mode reduces each of the injection period, the air-fuel mixture formation period, and the combustion period. The high pressure retarded injection performed in the range (II) in the CI mode reduces each of the injection period, and the air-fuel mixture formation period. Specifically, the turbulence in the cylinder **18** increases by injecting the fuel in the cylinder **18** at the high fuel pressure. This promotes mixing of the atomized fuel. Thus, even when the fuel is injected at the retarded timing near the compression top dead center, the relatively homogeneous air-fuel mixture can quickly be formed in the cylinder **18**.

The high pressure retarded injection in the CI mode is performed in the operation range of the engine where the load is relatively high. Specifically, the injector **67** injects the fuel in the cylinder **18** at the retarded timing near the compression top dead center. Thus, the preignition in the compression stroke can be prevented, and the nearly homogeneous air-fuel mixture can quickly be formed as described above. As a result, the air-fuel mixture is surely compression-ignited after the compression top dead center. As the combustion is performed in the expansion stroke where the pressure in the cylinder **18** gradually decreases due to motoring, the combustion proceeds slowly, thereby avoiding rapid increase in rate of pressure increase ($dP/d\theta$) in the cylinder **18** accompanying the compression ignition combustion. This resolves constraints of NVH, thereby expanding the range in the CI mode toward the high load range.

Back to the description of the SI mode, as described above, the high pressure retarded injection in the SI mode reduces the reactive time of the unburnt air-fuel mixture by injecting the fuel in the retarded period. The reduction of the reactive time is advantageous in the low speed range where the rotation speed of the engine **1** is relatively low because long actual time is required for changing a crank angle. In contrast, the reduction of the reactive time is less effective for avoiding the abnormal combustion in the high speed range where the rotation speed of the engine **1** is relatively high because the actual time required for changing the crank angle is short. Conversely, in the retarded injection, the fuel is injected in the cylinder **18** at timing near the compression top dead center. Thus, a gas in the cylinder free from the fuel, i.e., air having a high specific heat ratio, is compressed in the compression stroke. This increases the compression end temperature in the cylinder **18**, and the increased compression end temperature may lead to the knocking. Thus, in the retarded injection in the range (IV) where an amount of the fuel to be injected increases, the ignition timing needs to be retarded to avoid the knocking.

In the range (IV) in the SI mode where the rotation speed is relatively high, and the load is high as shown in FIG. **4**, the injector **67** injects part of the fuel in the cylinder **18** in the intake stroke, and injects the remaining fuel in the cylinder **18** in the retarded period as shown in FIG. **5**D. The injection in the intake stroke reduces the specific heat ratio of the gas in the cylinder (i.e., the air-fuel mixture containing the fuel) in the compression stroke, thereby reducing the compression end temperature. The reduced compression end temperature reduces the knocking, and the PCM **10** can advance the ignition timing.

The high pressure retarded injection in the range (IV) increases the turbulence in the cylinder **18** near the compression top dead center to reduce the combustion period as described above. This is also advantageous in reducing the knocking, and the PCM **10** can further advance the ignition timing. Thus, in the range (IV), the split injection including the injection in the intake stroke and the high pressure retarded injection avoids the abnormal combustion, and improves the thermal efficiency.

In the range (IV), the high pressure retarded injection may be replaced with multi-point ignition to reduce the combustion period. The multi-point ignition uses a plurality of spark plugs disposed in the combustion chamber. Specifically, in the range (IV), the injector **67** injects the fuel in the intake stroke, and the PCM **10** drives the plurality of spark plugs to perform the multi-point ignition. In the multi-point ignition, flame spreads from a plurality of sparks generated in the cylinder **18**, the flame rapidly grows to reduce the combustion period. As a result, like the high pressure retarded injection, the multi-point ignition reduces the combustion period to advantageously improve the thermal efficiency.

(Control of Switching from SI Mode to CI Mode)

In the spark ignition combustion, the thermal efficiency is low as compared with that of the compression ignition combustion, and the temperature of the combustion gas is relatively high. On the other hand, in the CI mode in which the compression ignition combustion is performed, at least the internal EGR gas is introduced into the cylinder **18** to increase the temperature in the cylinder **18** and to ensure the ignitability of the compression ignition as described above.

Immediately after the engine **1** is switched from the SI mode where the temperature of the combustion gas is relatively high to the CI mode, the temperature in the cylinder **18** is high, and the high temperature exhaust gas generated by the spark ignition combustion is introduced into the cylinder **18**. Thus, the engine **1** performs the compression ignition combustion while the temperature in the cylinder **18** is high. At this time, if the injector **67** injects the fuel in the cylinder **18** at relatively early timing, such as in the intake stroke, the preignition occurs in the compression stroke, thereby rapidly increasing the rate of pressure increase ($dP/d\theta$) in the cylinder **18**, and causing loud combustion noise. Thus, the engine **1** performs transitional control to avoid the preignition in switching from the SI mode to the CI mode, and to avoid the increase in combustion noise.

In the operation range in the hot state shown in FIG. **4**, for example, the switching from the SI mode to the CI mode is performed when the engine **1** is shifted from the high load range which is the SI mode to the low load range in which is the CI mode. Specifically, as the load of the engine **1** decreases, the engine **1** is switched from the SI mode to the CI mode. The engine **1** may be switched from the SI mode to the CI mode near the boundary between the SI mode and the CI mode even when the load of the engine **1** is unchanged.

When the engine **1** is in a cold or semi-warming state where the temperature of the engine **1** is lower than a predetermined temperature, the compression ignition combustion is not stabilized. Thus, although not shown, the PCM **10** does not perform the CI mode operation, and performs the SI mode operation in the whole operation range of the engine **1**. As shown in FIG. **4**, when the engine **1** is in the warm state where the temperature is at the predetermined temperature or higher, the PCM **10** performs the CI mode operation in part of the operation range. Thus, as the temperature gradually increases from the cold state to the warm state after the start of the engine **1**, the engine **1** may be switched from the SI mode to the CI mode with the load of the engine unchanged.

In view of stabilized combustion, the PCM **10** performs the SI mode operation when the engine **1** is idling. Thus, when the engine **1** is shifted from the idle state to the low load range where the CI mode operation is performed, the engine **1** is switched from the low load SI mode to the low load CI mode. In addition, the engine **1** is configured to stop the fuel supply when the vehicle is slowing down. Since the temperature in the cylinder **18** decreases while the fuel supply is stopped, the compression ignition combustion cannot be performed in some cases immediately after the fuel is supplied again. Thus, the PCM **10** performs the SI mode operation even in the range of the CI mode immediately after the fuel is supplied again to surely stabilize the combustion. Thereafter, as the temperature in the cylinder **18** increases so that the CI mode operation can be performed, the PCM **10** switches the SI mode to the CI mode. Specifically, the engine **1** is switched from the SI mode to the CI mode even though the load is not substantially changed. The switching from the low load SI mode to the low load CI mode, i.e., switching from the SI mode to the CI mode without substantial change in load of the engine **1**, is performed in various situations.

FIG. **7** is a timing chart illustrating the transitional control performed when the engine **1** is switched from the SI mode where the external EGR gas is not introduced to the CI mode. Specifically, FIG. **7** shows examples of how the timing of the fuel injection and the timing of the spark ignition are changed, how the pressure in the cylinder is changed, when and how the intake/exhaust valves are opened, how the degree of opening of the throttle valve is changed, and how the gas state in the cylinder is changed. In FIG. **7**, the crank angle varies from the left to the right of the paper (i.e., time progresses). The timing of the fuel injection, the timing of the spark ignition, and the change in pressure in the cylinder are described to illustrate the disclosed technology, and the disclosed technology is not limited to these timings (the same is applied to FIG. **8**, etc.). The SI mode in which the external EGR gas is not introduced corresponds to the SI mode operation performed when the load of the engine is low, for example. Thus, the timing chart shown in FIG. **7** corresponds to the switching of the engine **1** from the low load SI mode to the low load CI mode.

In the first cycle on the leftmost column in FIG. **7**, the engine **1** is operated in the SI mode. The injector **67** injects the fuel in a period from the intake stroke to the initial stage of the compression stroke, and the spark plug **25** performs spark ignition near the compression top dead center. In the first cycle, the air-fuel mixture has the theoretical air-fuel ratio ($\lambda \approx 1$), and the intake-side VVL **72** drives the intake valve **21** with the large lift cam, while the VVT **72** closes the intake valve **21** at retarded timing after the intake bottom dead center so that the amount of the fresh air is appropriate to the amount of the injected fuel. Thus, the PCM **10** retards the timing of closing the intake valve **21** to reduce the amount of the fresh air introduced into the cylinder **18** (see the gas state in the cylinder shown in the lowermost row in FIG. **7**). In the control example shown in FIG. **7**, the PCM **10** reduces the degree of opening of the throttle valve **36** in the first cycle to further reduce the amount of the fresh air to correspond to the operation state of the engine **1**. In preparation for the switching to the CI mode in which the throttle valve is fully opened, the PCM **10** gradually opens the throttle valve **36** to the full open state. As described above, the load of the engine **1** is low in the first cycle, although the first cycle is performed in the SI mode. Thus, the PCM **10** does not introduce the external EGR gas into the cylinder **18**. The exhaust-side VVL **71** is off. Specifically, the PCM **10** does not introduce the internal EGR gas into the cylinder **18**. In the first cycle in which the spark ignition combustion is performed, the temperature of the exhaust gas increases (i.e., the high temperature burnt gas is generated). Since the external and internal EGR gases are not introduced into the cylinder **18**, the high temperature exhaust gas is not substantially introduced into the cylinder **18** in the subsequent second cycle.

The second cycle corresponds to a cycle for switching from the SI mode to the CI mode. Specifically, this cycle is performed in a transitional mode. The PCM **10** makes the spark plug **25** inactive to perform the compression ignition combustion. Further, the PCM **10** fully opens the throttle valve, and the intake-side VVL **73** switches the large lift cam driving the intake valve to the small lift cam. The VVT **72** is inactive, and the phase of the intake valve **21** does not change. Thus, the timing of closing the intake valve **21** is instantaneously changed from the timing in the compression stroke in the first cycle to the timing near the intake bottom dead center, thereby increasing the amount of the fresh air introduced into the cylinder **18**. The timing of opening/closing the intake valve **21** corresponds to the timing of opening the exhaust valve twice started from the third cycle described later.

In the second cycle, the PCM **10** sets the timing of the fuel injection in the period from the intake stroke to the initial stage of the compression stroke as set in the first cycle, and sets the amount of the injected fuel similar to the amount injected in the first cycle. As compared with the first cycle, the amount of the fresh air introduced into the cylinder **18** increases, while the amount of the injected fuel does not change. Thus, in the second cycle, the air-fuel mixture has the A/F lean of the theoretical air-fuel ratio. Also in the second cycle, the PCM **10** does not introduce the exhaust gas (the external and internal EGR gases) into the cylinder **18**.

In the second cycle, the compression ignition combustion is performed, and the exhaust gas is not introduced into the cylinder **18**. Thus, the temperature in the cylinder **18** decreases. In the second cycle, the amount of the fresh air introduced into the cylinder **18** is increased to fill the cylinder **18** with the fresh air at relatively low temperature. This reduces the temperature in the cylinder **18** before the start of the compression. As a result, in the second cycle subsequent to the first cycle performed in the SI mode, and in which the temperature in the cylinder **18** relatively easily increases, the air-fuel mixture in the cylinder **18** does not cause the preignition, and is compression-ignited at appropriate timing near the compression top dead center. In the second cycle, the temperature of the combustion gas greatly decreases because the air-fuel mixture has the A/F ratio lean of the theoretical air-fuel ratio, the amount of the gas is large relative to the amount of the fuel, and the compression ignition combustion with high thermal efficiency is performed. This reduces the temperature of the exhaust gas emitted from the cylinder **18** in the second cycle. In the gas state shown in the lowermost row in FIG. **7**, the level of the temperature of the "burnt gas" is indicated by a pitch between hatch lines. A small pitch indicates the high temperature of the burnt gas, and a large pitch indicates the low temperature of the burnt gas.

The subsequent third cycle corresponds to a cycle for switching from the transitional mode to the CI mode. In the third cycle, the PCM **10** turns the exhaust-side VVL **71** on. Specifically, the exhaust valve **22** is opened twice, thereby introducing part of the burnt gas generated by the compression ignition combustion in the second cycle into the cylinder **18**.

In the third cycle, the intake-side VVL **73** opens/closes the intake valve **21** with the small lift cam, and the PCM **10** fully opens the throttle valve **36** in the same manner as in the second cycle. Thus, as shown in FIG. **7**, the internal EGR gas is introduced into the cylinder **18**, and the amount of the fresh air introduced into the cylinder **18** is reduced to be smaller than the amount introduced in the second cycle. Accordingly, an EGR ratio in the second cycle in the transitional mode is lower than the EGR ratio in the third cycle in the CI mode. The EGR ratio is a ratio of the amount of the exhaust gas to the whole amount of the gas in the cylinder **18**.

In the third cycle, part of the burnt gas is introduced into the cylinder **18**. However, the burnt gas is relatively low in temperature as described above. The compression ignition combustion performed in the second cycle reduces the temperature in the cylinder **18**. Thus, the temperature in the cylinder **18** in the third cycle does not excessively increase.

The third cycle corresponds to the normal CI mode. Thus, the PCM **10** sets the timing when the injector **67** injects the fuel in the period from the intake stroke to the initial stage of the compression stroke in the same manner as the second cycle. This allows formation of the relatively homogeneous air-fuel mixture in the cylinder **18**. The PCM **10** sets the amount of the fuel injected in the third cycle to be similar to the amount injected in the second cycle. This makes the A/F of the air-fuel mixture equal to or lean of the theoretical air-fuel ratio. The A/F of the air-fuel mixture is set in accordance with the operation state of the engine **1**.

The PCM **10** makes the spark plug **25** inactive in the third cycle in the same manner as the second cycle. As described above, the temperature in the cylinder **18** does not excessively increase in the third cycle, the preignition of the relatively homogeneous air-fuel mixture formed in the cylinder **18** does not occur, and the air-fuel mixture is surely compression-ignited near the compression top dead center, and stably combusted.

The transitional control avoids the increase in combustion noise in switching from the SI mode to the CI mode. After the third cycle in which the switching from the SI mode to the CI mode has been done, combustion control is performed in accordance with the operation state of the engine **1**.

FIG. **8** is a timing chart showing the transitional control in switching the engine **1** from the SI mode in which the external EGR gas is introduced to the CI mode. The timing chart of FIG. **8** corresponds to a situation in which the engine **1** is switched from the high load SI mode to the low load CI mode. Specifically, in the operation range in the warm state shown in FIG. **4**, the timing chart corresponds to a situation in which the engine **1** is shifted from the range (III) or the range (IV) where the engine **1** is operated in the SI mode to the range (I) or the range (II) where the engine is operated in the CI mode (see an arrow (a) in FIG. **4**).

Specifically, in the first cycle shown in the leftmost column of FIG. **8**, the engine **1** is operated in the SI mode. The injector **67** injects the fuel in the period from the terminal stage of the compression stroke to the initial stage of the expansion stroke. That is, the injector **67** performs the high pressure retarded injection. The spark plug **25** performs the spark ignition near the compression top dead center. The air-fuel mixture has the theoretical air-fuel ratio ($\lambda \approx 1$). The intake-side VVL **73** drives the intake valve **21** with the large lift cam, and the VVT **72** sets the timing of closing the intake valve **21** to timing after the intake bottom dead center so that the amount of the fresh air is appropriate to the amount of the injected fuel. The PCM **10** retards the timing of closing the intake valve **21** to reduce the amount of the fresh air introduced into the cylinder **18**. In the first cycle in the timing chart of FIG. **8**, the PCM **10** reduces the degree of opening of the throttle valve **36**, and gradually increases the degree of opening to the full open state in the same manner as in the first cycle in the timing chart of FIG. **7** described above.

In the first cycle in the timing chart of FIG. **8**, the PCM **10** opens the EGR valve **511** and/or the EGR cooler bypass valve **531**. Thus, the external EGR gas is introduced into the cylinder **18** through the EGR passage **50** as described above. The exhaust-side VVL **71** is off, and the internal EGR gas is not introduced into the cylinder **18**. The temperature of the exhaust gas may increase in the first cycle where the spark ignition combustion is performed.

The subsequent second cycle corresponds to the transitional mode for switching from the SI mode to the CI mode. Thus, in the second cycle, the PCM **10** makes the spark plug **25** inactive to perform the compression ignition combustion. The PCM **10** fully opens the throttle valve, and the intake-side VVL **73** switches the large lift cam driving the intake valve to the small lift cam. Thus, the timing of closing the intake valve **21** is instantaneously changed from the timing in the compression stroke in the first cycle to the timing near the intake bottom dead center, thereby increasing the amount of the fresh air introduced into the cylinder **18**.

In the second cycle, the PCM **10** completely closes the EGR valve **511** and/or the EGR cooler bypass valve **531**. This stops the introduction of the external EGR gas into the cylinder **18**. However, since the external EGR system does not show a quick response to the control, the relatively hot exhaust gas remaining in the EGR passage **50** is introduced into the cylinder **18** in the second cycle even after the EGR valve **511** and the EGR cooler bypass valve **531** are fully closed (see the gas state shown in the lowermost row in FIG. **8**). On the other hand, the exhaust-side VVL **71** is still off in the second cycle. Specifically, the PCM **10** does not introduce the internal EGR gas into the cylinder **18**.

The PCM **10** sets the amount of the fuel injected by the injector **67** to be substantially the same as the amount injected in the first cycle. The PCM **10** sets the timing of injecting the fuel in the period from the intake stroke to the initial stage of the compression stroke. Thus, the relatively homogeneous air-fuel mixture which is lean of the theoretical air-fuel ratio is formed in the cylinder **18** in the second cycle.

The PCM **10** does not activate the spark plug **25** in the second cycle, and allows compression auto-ignition of the homogeneous air-fuel mixture lean of the theoretical air-fuel ratio near the compression top dead center. In the second cycle, the amount of the fresh air introduced into the cylinder **18** increases to decrease the temperature in the cylinder **18** before the start of the compression. As a result, the air-fuel mixture in the cylinder **18** is compression-ignited at appropriate timing near the compression top dead center without causing the preignition. In the second cycle, the A/F of the air-fuel mixture is lean of the theoretical air-fuel ratio, and a ratio of a working gas to the fuel G/F is lean. Further, the temperature of the combustion gas decreases because the compression ignition combustion with high thermal efficiency is performed. As a result, the temperature of the exhaust gas emitted in the second cycle decreases.

As described above, part of the external EGR gas is introduced into the cylinder **18** in the second cycle. Accordingly, the temperature of the exhaust gas may slightly increase. For convenience's sake, the burnt gas in the second cycle is referred to as an intermediate temperature burnt gas at an intermediate temperature between the temperature of a high temperature burnt gas and the temperature of a low temperature burnt gas. If the exhaust-side VVL **73** is turned on in the subsequent third cycle to introduce a large amount of the intermediate temperature burnt gas into the cylinder **18**, the temperature in the cylinder **18** may excessively increase.

Thus, in the control example shown in FIG. **8**, the PCM **10** performs several cycles continuously in the transitional mode. Specifically, the PCM **10** turns the exhaust side VVL **71** off to perform the third cycle in the same manner as the second cycle. In the third cycle, the exhaust gas does not remain in the EGR passage **50**, and the exhaust gas is not introduced into the cylinder **18**. As a result, the temperature in the cylinder **18** before the start of the compression is lower than the temperature in the second cycle, and the A/F of the air-fuel mixture is lean of the A/F in the second cycle. Thus, the temperature of the exhaust gas emitted through the compression ignition combustion in the third cycle is reduced to be lower than the temperature of the exhaust gas emitted in the second cycle (i.e., the low temperature burnt gas is emitted).

The fourth cycle corresponds to a cycle for switching from the transitional mode to the CI mode. The fourth cycle in the timing chart of FIG. **8** corresponds to the third cycle in the timing chart of FIG. **7**. The PCM **10** turns the exhaust-side VVL **71** on to open the exhaust valve **22** twice. Thus, part of the relatively low temperature burnt gas generated by the compression ignition combustion in the third cycle is introduced into the cylinder **18**. In the fourth cycle, the amount of the fresh air introduced into the cylinder **18** is reduced as compared with the amount of the fresh air introduced in the second and third cycles because the internal EGR gas is introduced into the cylinder **18**. Specifically, the EGR ratio relatively increases in the fourth cycle.

Since the fourth cycle corresponds to the normal CI mode, the injector **67** injects the fuel in the period from the intake stroke to the initial stage of the compression stroke. The amount of the fuel injected by the injector **67** is substantially the same as the amount injected in the second and third cycles in the transitional mode. The A/F of the air-fuel mixture is rich as compared with the A/F in the second and third cycles, and is theoretical, or lean of the theoretical air-fuel ratio.

Also in the fourth cycle, the PCM **10** makes the spark plug **25** inactive to perform the compression ignition combustion. Since the temperature in the cylinder **18** does not excessively increase as described above, the preignition of the relatively homogeneous air-fuel mixture does not occur, and the air-fuel mixture is surely compression-ignited near the compression top dead center, and stably combusted.

When the engine **1** is operated under a high load, the injector **67** injects a large amount of the fuel. Thus, when the engine **1** is operated in the high load SI mode, the temperature in the cylinder **18** increases, and the temperature of the exhaust gas increases. Accordingly, the preignition easily occurs particularly in switching from the high load SI mode to the low load CI mode. In the control example of FIG. **8**, two cycles are continuously performed in the transitional mode. This is advantageous in surely avoiding the preignition in switching from the SI mode to the CI mode.

In the transitional mode, the EGR ratio in the cylinder **18** is reduced as compared with the EGR ratio in the CI mode to reduce the temperature in the cylinder **18**, thereby avoiding the preignition. However, as described above, the temperature in the cylinder **18** increases particularly in the high load SI mode. When the temperature of a wall of the cylinder **18** is high, the preignition may not be avoided in the transitional mode. In such a case, the PCM **10** may retard the timing of the fuel injection by the injector **67** after the intermediate stage of the compression stroke as indicated by a broken line in FIG. **8**. Specifically, the PCM **10** may perform the retarded injection in the transitional mode. In the retarded injection, the fuel is injected at retarded timing, and the reactive time of the air-fuel mixture is reduced. Thus, the preignition in the transitional mode is advantageously avoided. The PCM **10** may estimate the temperature in the cylinder **18** based on various parameters, for example, and may set the timing of the fuel injection by the injector **67** in the period from the intake stroke to the initial stage of the compression stroke, or after the intermediate stage of the compression stroke based on the estimation results.

The retarded injection is not performed only in the transitional mode. The PCM **10** may perform the retarded injection as needed after the fourth cycle in which the engine **1** is switched to the CI mode as indicated by a dot-and-dash line in FIG. **8**. Specifically, the PCM **10** switches the engine **1** from the transitional mode to the CI mode when the ignitability is not ensured unless the exhaust-side VVL **71** is turned on to introduce the internal EGR gas into the cylinder **18**, while the PCM **10** allows the injector **67** to perform the retarded injection when the preignition may possibly occur in the CI mode. This makes it possible to surely avoid the generation of the combustion noise in switching the engine **1** from the SI mode to the CI mode, and to stably perform the compression ignition combustion. In switching from the low load SI mode to the low load CI mode as shown in FIG. **7**, the PCM **10** may perform the retarded injection as needed in the transitional mode or the CI mode.

Referring to FIG. **9**, a flow of the above-described transitional control performed by the PCM **10** will be described below. The flow of FIG. **9** starts in the SI mode where the excess air ratio $\lambda \approx 1$.

In step S**91** after the start, the PCM **10** reads various parameters (e.g., a water temperature, an outside air temperature, the load of the engine, the rotational speed of the engine, the timing of the fuel injection, the fuel pressure, the timing of ignition, the timing of opening/closing the intake valve, the timing of opening/closing the exhaust valve, etc.), and check the operation state of the engine **1**. In step S**92**, the PCM **10** determines whether the engine **1** is switched from the SI mode to the CI mode or not. When the engine **1** is not switched to the CI mode (NO is chosen), the flow repeats the steps S**91** and S**92**. When the engine is switched to the CI mode (YES is chosen), the flow proceeds to step S**93**. Specifically, in step S**92**, YES is chosen when the engine **1** operated in the high load state is changed to the low load state, when the engine **1** in the cold state is changed to the warm state, when the idling engine **1** is changed to a low load operation state except for the idling state, and when the engine **1** temporarily operated in the SI mode after the restart of the fuel supply is switched to the CI mode, for example. A period from the start of the flow to step S**92** corresponds to the first cycle in the timing chart of FIGS. **7** and **8**.

In step S**93**, the PCM **10** determines whether the external EGR gas is introduced into the cylinder **18** or not. The flow proceeds to step S**94** when the external EGR gas is introduced into the cylinder **18**. The PCM **10** completely closes the EGR valve **511** and/or the EGR cooler bypass valve **531** in step S**94**. The flow proceeds from step S**93** to step S**95** when the external EGR gas is not introduced into the cylinder **18**.

In step S**95**, the PCM **10** switches the large lift cam of the intake-side VVL **73** driving the intake valve to the small lift cam. In subsequent step **S96**, the PCM **10** determines whether the cam of the intake-side VVL **71** has been switched or not. When the cam has not been switched (NO is chosen), the flow continues step **S95**. When the cam has been switched (YES is chosen), the flow proceeds to step **S97**. The cam is instantaneously switched. In step **S97**, the PCM **10** makes the spark plug **25** inactive, thereby switching the engine **1** from the SI mode to the CI mode.

In subsequent step **S98**, the PCM **10** determines whether the state in the cylinder is ready or not, i.e., whether the temperature of the exhaust gas has been reduced to a desired level or not. For example, the PCM **10** may perform the determination in this step based on the temperature of the exhaust gas estimated based on the various parameters. The flow continues step **S98** when the result of the determination is NO in step **S98**. The flow proceeds to step **S99** when the result of the determination is YES. Steps **S93**-**S98** correspond to the second cycle in the timing chart of FIG. **7** and the second and third cycles in the timing chart of FIG. **8** in the transitional mode. Thus, the number of cycles performed in the transitional mode is determined in step **S98**. Specifically, while only a single cycle, i.e., the second cycle, is performed in the transitional mode in the control example of FIG. **7**, the PCM **10** may perform several cycles continuously in the transitional mode based on the temperature of the exhaust gas. In the control example of FIG. **8**, two cycles, i.e., the second and third cycles, are performed in the transitional mode. The PCM **10** may perform three or more cycles, or only a single cycle, in the transitional mode based on the temperature of the exhaust gas.

In step **S98**, the PCM **10** determines the number of cycles performed in the transitional mode based on the estimated temperature of the exhaust gas. Alternatively, the PCM **10** may previously store the number of cycles performed in the transitional mode based on the operation state of the engine **1** before and after the switching from the SI mode to the CI mode, etc. so that the PCM **10** performs the determined number of cycles in the transitional mode in step **S98**.

In step **S99** after step **S98**, the PCM **10** turns the exhaust-side VVL **71** on to start opening the exhaust valve **22** twice. This step corresponds to the third cycle shown in FIG. **7**, and the fourth cycle shown in FIG. **8**. Thus, the engine **1** is switched from the SI mode to the CI mode.

(Details of Valve Control)

Details of control of the intake valve **21** and the exhaust valve **22** in the switching from the SI mode to the CI mode will be described with reference to FIG. **10**. In the first cycle in the timing chart of FIG. **7** or FIG. **8**, i.e., in the SI mode, the exhaust valve **22** is opened in the exhaust stroke as indicated by a dot-and-dash line in FIG. **10**, and the intake valve **21** is opened in the intake stroke as indicated by a broken line in FIG. **10**. Specifically, the exhaust-side VVL **71** is operated in the normal mode, and the intake-side VVL **73** drives the intake valve with the large lift cam. This corresponds to a first valve operation.

In the transitional mode (i.e., in the second cycle shown in FIG. **7** and the second and third cycles shown in FIG. **8**), the exhaust valve **22** keeps showing a lift characteristic indicated by the dot-and-dash line. The intake valve **21** is opened for a shorter period as illustrated by a thin solid line as compared with the opening period in the first valve operation. Thus, in the transitional mode, the exhaust valve **22** is opened in the exhaust stroke, while the intake valve **21** is opened in the intake stroke like in the SI mode. In switching from the SI mode to the transitional mode, the cam of the exhaust-side VVL **71** is not switched, while only the large lift cam of the intake-side VVL **73** is switched to the small lift cam. The lift characteristic of the intake valve **21** can instantaneously be changed by switching the cam of the intake-side VVL. This corresponds to a second valve operation.

In the third cycle shown in FIG. **7** and the fourth cycle shown in FIG. **8** corresponding to the CI mode, the exhaust valve **22** is opened twice, and shows the lift characteristic as indicated by a thick solid line in FIG. **10**. The intake valve **21** keeps showing the lift characteristic indicated by the thin solid line. Specifically, in switching from the transitional mode to the CI mode, the cam of the intake-side VVL **73** is not switched, and only the cam of the exhaust-side VVL **71** is switched. This corresponds to a third valve operation. In the third valve operation, in addition to the second valve operation, the exhaust valve **22** is brought into an open state at timing later than the timing of closing the exhaust valve **22** in the second valve operation.

The configuration including the intake-side VVL **73** and the exhaust-side VVL **71** improves a response to the control for switching from the SI mode to the CI mode through the transitional mode, and is advantageous in smoothly switching from the SI mode to the CI mode.

In opening the exhaust valve **22** twice, the lift characteristic shown in FIG. **10** may be replaced with lift characteristics shown in FIGS. **11** and **12**. The lift characteristic of the exhaust valve **22** illustrated by the thick solid line in FIG. **10** indicates that the lift amount of the exhaust valve **22** maximized in the exhaust stroke is gradually reduced, and then increased again in the intake stroke. Specifically, the lift characteristic has two peaks of the lift amount of the exhaust valve **22** as the crank angle varies. The lift characteristic of the exhaust valve **22** indicates that the exhaust valve **22** is in the open state at timing later than the timing of closing the exhaust valve **22** in the SI mode as compared with the lift characteristic of the exhaust valve **22** in the SI mode illustrated by the dot-and-dash line in FIG. **10**.

The lift characteristic shown in FIG. **11** indicates that the lift amount of the exhaust valve **22** maximized in the exhaust stroke is gradually reduced to a predetermined level, and is kept at the predetermined level until predetermined timing in the intake stroke. Specifically, the lift characteristic shows a substantially single peak, while a bottom part of the characteristic curve extends in a direction of the variation of the crank angle. Although not shown, after the lift amount of the exhaust valve **22** maximized in the exhaust stroke is gradually reduced to the predetermined level, the rate of decrease of the lift amount is not reduced to zero, but may be reduced to a lower value to keep the exhaust valve **22** open for a longer time while reducing the degree of opening of the exhaust valve **22**. The lift characteristic shown in FIG. **11** also indicates that the exhaust valve **22** is in the open state at the timing later than the timing of closing the exhaust valve **22** in the SI mode.

The lift characteristics of the exhaust valve **22** shown in FIG. **10** and FIG. **11** indicate that the exhaust valve **22** is not closed near an exhaust top dead center. On the other hand, as indicated by the lift characteristic of FIG. **12**, the exhaust valve **22** opened twice may be once closed near the exhaust top dead center. This lift characteristic also indicates that the exhaust valve **22** is in the open state at the timing later than the timing of closing the exhaust valve **22** in the SI mode.

In the above-described control example, the PCM **10** opens the exhaust valve **22** twice in the CI mode. As described above, the engine **1** may be configured to open the intake valve **21** twice in place of opening the exhaust valve **22** twice. FIG. **13** shows the lift characteristics of the valves when the intake valve **21** is opened twice. The lift characteristic of the intake valve **21** opened twice can be line-symmetric to the lift characteristic of the exhaust valve **22** opened twice shown in FIG. **10** about the exhaust top dead center. The valve control in switching from the SI mode to the CI mode can be performed in a reverse order of the control for opening the exhaust valve **22** twice.

Specifically, in the cycle corresponding to the SI mode, the exhaust valve **22** is opened in the exhaust stroke as indicated by a dot-and-dash line in FIG. **13**, while the intake valve **21** is opened in the intake stroke as indicated by a broken line in FIG. **13**. This corresponds to the first valve operation. The intake-side VVL **73** operates the intake valve in the normal mode, while the exhaust-side VVL **71** operates the exhaust valve with the large lift cam.

In the cycle corresponding to the transitional mode, the intake valve **21** keeps showing the lift characteristic indicated by the broken line, while the exhaust valve **22** is opened for a shorter period than the opening period in the first valve operation as indicated by the thick solid line. Specifically, only the cam of the exhaust-side VVL **71** is switched, while the cam of the intake-side VVL **73** is not switched in switching from the SI mode to the transitional mode. This corresponds to the second valve operation.

In the cycle corresponding to the CI mode, the intake valve **21** is opened twice to show the lift characteristic indicated by the thin solid line in FIG. **10**. The exhaust valve **22** keeps showing the lift characteristic indicated by the thick solid line. Specifically, only the cam of the intake-side VVL **73** is switched, while the cam of the exhaust-side VVL **71** is not switched in switching from the transitional mode to the CI mode. Thus, in the CI mode, the intake valve **21** is in the open state at timing earlier than the timing of opening the intake valve **21** in the SI mode indicated by the broken line in FIG. **13**. This corresponds to the third valve operation.

The lift characteristic of the intake valve **21** opened twice is not limited to the lift characteristic shown in FIG. **13**. Although not shown, the intake valve **21** may show a lift characteristic which is line-symmetric to the lift characteristic shown in FIG. **11** about the exhaust top dead center, or a lift characteristic which is line-symmetric to the lift characteristic shown in FIG. **12** about the exhaust top dead center.

As described above, in switching the engine **1** from the SI mode to the CI mode as indicated by the arrow (a) in FIG. **4**, the engine **1** in the SI mode is once switched to the transitional mode in which the EGR ratio is relatively reduced before switching to the CI mode. This can avoid the preignition in the switching from the SI mode and the CI mode, and can avoid the generation of the combustion noise. In switching the engine **1** from the SI mode to the CI mode without substantial change in load of the engine **1**, the engine **1** in the SI mode is once switched to the transitional mode before switching to the CI mode.

As indicated by an arrow (b) in FIG. **4**, for example, the combustion noise does not occur in switching the engine **1** from the CI mode to the SI mode because the load of the engine **1** increases. Thus, the engine **1** is not operated in the transitional mode in switching from the CI mode to the SI mode so that the mode can quickly be changed. Specifically, in the timing chart of FIG. **7**, the flow returns from the third cycle to the first cycle without performing the second cycle in switching from the CI mode to the SI mode. Likewise, in the timing chart of FIG. **8**, the flow returns from the fourth cycle to the first cycle without performing the second and third cycles in switching from the CI mode to the SI mode. This improves a response to the driver's request for acceleration.

(Other Configuration of Engine)

In the configuration shown in FIG. **2**, the valve mechanism of the intake valve **21** includes the VVL **73** configured to perform the switching between the large lift cam and the small lift cam. Unlike this configuration, as shown in FIG. **14**, the valve mechanism of the intake valve **21** may include, in place of the VVL, a continuously variable valve lift (CVVL) **74** capable of continuously varying the lift amount of the intake valve. The CVVL **74** may have various known configurations as appropriate, and details thereof are not described. The VVT **72** and the CVVL **74** make it possible to continuously change the timing of opening/closing the intake valve **21**, and the lift amount of the intake valve **21** (and a period in which the valve is opened).

FIGS. **15** and **16** show examples of the switching from the SI mode to the CI mode using the valve mechanism of the intake valve **21** including the CVVL **74**. FIG. **15** shows a timing chart related to the switching from the SI mode in which the external EGR gas is not introduced to the CI mode, and corresponds to FIG. **7**. FIG. **16** shows a timing chart related to the switching from the SI mode in which the external EGR gas is introduced to the CI mode, and corresponds to FIG. **8**.

The first cycle in the leftmost column in FIG. **15** is substantially the same as the first cycle in FIG. **7**. Specifically, the engine **1** is operated in the SI mode. The injector **67** injects the fuel in the period from the intake stroke to the initial stage of the compression stroke, and the spark plug **25** performs the spark ignition near the compression top dead center. The air-fuel mixture has the theoretical air-fuel ratio ($\lambda \approx 1$).

The intake-side CVVL **74** drives the intake valve **21** with a relatively small lift amount to control the amount of the fresh air appropriate to the amount of the injected fuel, and the VVT **72** sets the timing of closing the intake valve **21** to relatively early timing before the intake bottom dead center. Thus, the intake valve **21** is closed earlier, and the amount of the fresh air introduced into the cylinder **18** is reduced. Also in the control example of FIG. **15**, the PCM **10** reduces the degree of opening of the throttle valve **36** in the first cycle. However, the PCM **10** gradually opens the throttle valve **36** to the full open state in preparation for the switching to the CI mode. Further, the PCM **10** closes the EGR valve **511** and/or the EGR cooler bypass valve **531**, and turns the exhaust-side VVL **71** off. Thus, the external EGR gas and the internal EGR gas are not introduced into the cylinder **18**.

The subsequent second cycle corresponds to a cycle immediately before the switching from the SI mode to the CI mode. In the second cycle, in preparation for opening the exhaust valve **22** twice after the switching to the CI mode, the PCM **10** activates the VVT **72** to retard the timing of closing the intake valve **21**. The PCM **10** continues the operation of opening the throttle valve **36**. Thus, the amount of the fresh air introduced into the cylinder **18** increases as compared with the amount introduced in the first cycle. The PCM **10** increases the amount of the injected fuel in the second cycle. The A/F of the air-fuel mixture is kept to the theoretical air-fuel ratio. This allows use of the three-way catalyst, and avoids deterioration of the exhaust emission performance.

The PCM **10** allows the injector **67** to perform the high pressure retarded injection in the second cycle. The injector **67** injects the fuel near the compression top dead center. The

PCM **10** allows the spark plug **25** to perform the spark ignition at timing after the compression top dead center. Thus, the timing of the ignition is retarded to retard the start of the combustion period. Thus, the load of the engine **1** is kept unchanged in the first and second cycles (the torque is kept unchanged). Retarding the start of the combustion period further increases the temperature of the exhaust gas (the burnt gas), i.e., a super high temperature burnt gas is generated.

The third cycle corresponds to a cycle performed in the transitional mode for switching from the SI mode from the CI mode. The PCM **10** makes the spark plug **25** inactive to perform the compression ignition combustion. The PCM **10** fully opens the throttle valve. The intake valve **21** and the exhaust valve **22** keep showing the same lift characteristics as those in the second cycle. Specifically, the PCM **10** keeps the exhaust-side VVL **71** in the off state. Thus, the high temperature burnt gas generated by the spark ignition combustion in the second cycle is not substantially introduced into the cylinder **18**. As a result, the temperature in the cylinder **18** is relatively lowered. In the third cycle, the PCM **10** may turn the exhaust-side VVL **71** on, and may reduce the amount of the internal EGR gas to be smaller than the amount of the internal EGR gas set in the CI mode.

In the third cycle, the PCM **10** allows the injector **67** to inject the fuel in the cylinder **18** in the period from the intake stroke to the initial stage of the compression stroke as previously set in the CI mode. Since the temperature in the cylinder **18** is relatively low, the preignition is avoided, and the homogeneous air-fuel mixture in the cylinder **18** is compression-ignited near the compression top dead center, and combusted. Since the temperature of the burnt gas in the second cycle is high, the temperature of the gas remaining in the cylinder **18** increases. As a result, excessive decrease of the temperature in the cylinder **18** is avoided without opening the exhaust valve **22** twice. Thus, the homogeneous air-fuel mixture is surely compression-ignited near the compression top dead center, and stably combusted. The temperature of the combustion gas decreases because the compression ignition combustion takes place in the third cycle, and the ratio of the working gas to the fuel G/F is lean.

The fourth cycle corresponds to the CI mode after the transitional mode is finished. This cycle is substantially the same as the fourth cycle shown in FIG. **7**. Specifically, the PCM **10** turns the exhaust-side VVL **71** on as set in the normal CI mode. Thus, part of the burnt gas generated by the compression ignition combustion in the third cycle is introduced into the cylinder **18**. As described above, the burnt gas is relatively low in temperature, and decreases the temperature in the cylinder **18**. In the fourth cycle, the PCM **10** sets the timing of the fuel injection in the period from the intake stroke to the initial stage of the compression stroke. The preignition is avoided, and the homogeneous air-fuel mixture in the cylinder **18** is compression-ignited near the compression top dead center. As described above, the PCM **10** may perform the retarded injection in the transitional mode or the CI mode.

In the first cycle in the leftmost column of the timing chart of FIG. **16**, the engine **1** is operated in the SI mode while introducing the external EGR gas into the cylinder. The air-fuel mixture has the theoretical air-fuel ratio ($\lambda \approx 1$). The PCM **10** allows the injector **67** to perform the retarded injection. The CVVL **74** of the intake valve **21** drives the intake valve **21** with a relatively small lift amount so that the amount of the fresh air is appropriate to the amount of the injected fuel. The VVT **72** closes the intake valve **21** at relatively early timing before the intake bottom dead center. The PCM **10** reduces the degree of opening of the throttle valve **36**. The PCM **10** introduces the external EGR gas into the cylinder **18** through the EGR passage **50**. The VVL **71** of the exhaust valve **22** is off.

The second cycle is the same as the second cycle in the timing chart of FIG. **15**. The PCM **10** continues the SI mode, and retards the timing of closing the intake valve **21** in preparation for opening the exhaust valve **22** twice. Retarding the timing of closing the intake valve increases the amount of the fresh air introduced into the cylinder **18**. The PCM **10** increases the amount of the injected fuel to correspond to the increased fresh air. Thus, the air-fuel mixture has the theoretical air-fuel ratio also in the second cycle. The injector **67** performs the retarded injection to inject the fuel into the cylinder **18**.

The third cycle corresponds to a cycle in the transitional mode for switching from the SI mode to the CI mode. The PCM **10** stops the operation of the spark plug **25**, and closes the EGR valve **511** and the EGR cooler bypass valve **531**. Part of the exhaust gas remaining in the EGR passage **50** is introduced into the cylinder **18**.

The PCM **10** reduces the amount of the fuel injected by the injector **67** to approach the amount of the fuel injected in the first cycle. As a result, the A/F of the air-fuel mixture is lean of the theoretical air-fuel ratio. The injector **67** may inject the fuel into the cylinder **18** in the period from the intake stroke to the initial stage of the compression stroke. As described above, when the temperature in the cylinder **18** is high, and the preignition may possibly occur, the PCM **10** may retard the timing of the fuel injection after the terminal stage of the compression stroke. Thus, in the third cycle, the homogeneous air-fuel mixture lean of the air-fuel mixture in the first and second cycles is combusted by the compression ignition. This decreases the temperature of the exhaust gas. However, as described above, part of the high temperature burnt gas is introduced into the cylinder **18** in the third cycle, although the PCM **10** closes the EGR valve **511** and the EGR cooler bypass valve **531**. As a result, the temperature of the exhaust gas is not sufficiently reduced. Thus, in the control example of FIG. **16**, the PCM **10** performs the subsequent fourth cycle in the transitional mode in the same manner as in the third cycle. The fourth cycle performed in the transitional mode sufficiently decreases the temperature of the exhaust gas.

In the subsequent fifth cycle, when the PCM **10** turns the exhaust-side VVL **71** on to start opening the exhaust valve **22** twice, the exhaust gas at the reduced temperature is introduced into the cylinder **18**. Thus, the compression ignition combustion can stably be performed without causing the preignition. In the fifth cycle corresponding to the CI mode, the switching from the SI mode to the CI mode is finished.

Thus, the valve mechanism of the intake valve **21** including the CVVL **74** can also perform the transitional control in switching from the SI mode from the CI mode. Note that the VVL **73** can instantaneously switch the SI mode to the transitional mode as described above. Therefore, as apparent from the comparison between FIG. **7** and FIG. **15**, or the comparison between FIG. **8** and FIG. **16**, the minimum number of cycles required for the mode switching is different. The VVL **73** is more advantageous because it improves the response to the transitional control, and smoothly performs the mode switching.

(Other Example of Operation Range)

FIG. **17** shows another example of the operation range of the engine **1** in the warm state. Unlike the operation range shown in FIG. **4**, the range in the CI mode is divided into three ranges in accordance with the level of the load of the engine in the operation range of FIG. **17**. Specifically, a range (i) where the load is the lowest in the CI mode corresponds to the range (I) of FIG. **4**. In the range (i), the hot EGR gas is introduced into the cylinder **18**. Further, in the range (i), the injector **67** injects the fuel in the cylinder **18** at least in the period from the intake stroke to the intermediate stage of the compression stroke as shown in FIG. **5**A. Thus, a homogeneous and lean air-fuel mixture is formed. The air-fuel mixture may have an excess air ratio λ of 2.4 or higher, for example. This can reduce the generation of raw NOx, and can improve the exhaust emission performance. In a high load range in the range (i), i.e., in a range including a boundary between the range (i) and a range (ii), the air-fuel mixture has the theoretical air-fuel ratio (λ≈1).

In the CI mode, the fuel is injected into the cylinder **18** at least in the period from the intake stroke to the intermediate stage of the compression stroke in the range (ii) where the load is higher than the load in the range (i), like in the high load range in the range (i) (see FIG. **5**A). Thus, the homogeneous air-fuel mixture having the theoretical air-fuel ratio (λ≈1) is formed in the cylinder **18**.

In the range (ii), the high temperature hot EGR gas and the low temperature cooled EGR gas are introduced into the cylinder **18** in an appropriate ratio. This optimizes the compression end temperature in the cylinder **18**, and stabilizes the compression ignition combustion while ensuring the ignitability of the compression ignition, and avoiding rapid combustion. An EGR ratio, which is a ratio of an EGR gas, i.e., a mixture of the hot EGR gas and the cooled EGR gas introduced into the cylinder **18**, is set as high as possible under the condition that the A/F of the air-fuel mixture λ≈1. Therefore, in the range (ii), the EGR ratio gradually decreases because the amount of the injected fuel increases with the increase in load of the engine.

A range (iii) where the load is the highest in the CI mode and a boundary between the CI mode and the SI mode exists corresponds to the range (II) in the operation range shown in FIG. **4**. In this range, the temperature in the cylinder **18** is controlled, and the high pressure retarded injection is performed.

While the operation range in the CI mode is divided into the three ranges in accordance with the level of the load of the engine, an operation range in the SI mode is divided into two ranges, i.e., a range (iv) and a range (v), in accordance with the level of the engine speed. In the illustrated example, suppose that the operation range of the engine **1** is divided into a low speed range and a high speed range, the range (iv) corresponds to the low speed range, and the range (v) corresponds to the high speed range. A boundary between the range (iv) and the range (v) is tilted in the direction of the engine speed relative to the level of the load in the operation range shown in FIG. **17**. However, the boundary is not limited to the illustrated boundary.

The range (iv) corresponds to the range (IV) in FIG. **4**. The injector **67** performs the retarded injection (see FIG. **5**C). The range (v) corresponds to the range (V) in FIG. **4**. The injector **67** performs the split injection including the retarded injection (see FIG. **5**D).

In this operation range in the warm state, the above-described transitional control may be performed in switching from the range (iv) or the range (v) in which the engine **1** is operated in the SI mode to the range (i), the range (ii), or the range (iii) in which the engine **1** is operated in the CI mode, for example.

Other Embodiments

The disclosed technology is not limited to the application to the engine configured as described above. For example, in the intake stroke, the fuel injection from the injector **67** disposed in the cylinder **18** may be replaced with the fuel injection into the intake port **16** through a port injector provided in the intake port **16**.

The engine **1** is not limited to the in-line four-cylinder engine, and may be an in-line three-cylinder engine, an in-line two-cylinder engine, an in-line six-cylinder engine, etc. The disclosed technology is applicable to various types of engines, such as a V-type six-cylinder engine, a V-type eight-cylinder engine, a horizontally opposed four cylinder engine, etc.

In the above description, the air-fuel mixture has the theoretical air-fuel ratio (λ≈1) in the predetermined operation range. However, the air-fuel mixture may have a lean air-fuel ratio. Note that setting the air-fuel ratio to the theoretical air-fuel ratio is advantageous because the three-way catalyst can be used.

The high pressure retarded injection may be replaced with the split injection as needed. Likewise, the injection in the intake stroke may be replaced with the split injection as needed. In the split injection, the fuel may be injected in each of the intake stroke and the compression stroke.

DESCRIPTION OF REFERENCE CHARACTERS

1 Engine (engine body)
10 PCM (controller)
18 Cylinder
21 Intake valve
22 Exhaust valve
25 Spark plug
50 EGR passage (exhaust returning apparatus)
51 Main passage (exhaust returning apparatus)
511 EGR valve (exhaust returning apparatus)
52 EGR cooler (exhaust returning apparatus)
53 EGR cooler bypass passage (exhaust returning apparatus)
531 EGR cooler bypass valve (exhaust returning apparatus)
67 Injector (fuel injection valve)
71 VVL (exhaust returning apparatus, internal EGR regulator)
72 VVT (valve mechanism)
73 VVL (valve mechanism)
74 CVVL (valve mechanism)

The invention claimed is:

1. A spark ignition engine comprising:

an engine body having a cylinder;

a spark plug disposed to face an inside of the cylinder, and configured to ignite an air-fuel mixture in the cylinder;

an exhaust returning apparatus configured to introduce an exhaust gas into the cylinder; and

a controller configured to operate the engine body by controlling at least the spark plug and the exhaust returning apparatus, wherein

the controller performs switching between a compression ignition mode in which compression ignition combustion is performed by auto-ignition of the air-fuel mixture in the cylinder to operate the engine body, and a spark ignition mode in which spark ignition combustion is performed by driving the spark plug to ignite and combust the air-fuel mixture in the cylinder to operate the engine body,

the controller controls the exhaust returning apparatus at least in the compression ignition mode to introduce the exhaust gas into the cylinder so that an EGR ratio which is a ratio between an amount of the exhaust gas to a total amount of a gas in the cylinder is a predetermined value, and the controller reduces an amount of the exhaust gas introduced into the cylinder and increases an amount of fresh air introduced in the cylinder to reduce the EGR ratio to be lower than the EGR ratio set in the compression ignition mode, and to make an air-fuel ratio of the air-fuel mixture in the cylinder lean of a theoretical air-fuel ratio and lean of the air-fuel ratio in the spark ignition mode before switching the engine body to the compression ignition mode, and makes the ignition plug inactive, thereby switching the engine body operated in the spark ignition mode to a transitional mode in which the compression ignition combustion is performed before switching to the compression ignition mode.

2. The spark ignition engine of claim **1**, wherein

the exhaust returning apparatus includes an internal EGR regulator for regulating an amount of the exhaust gas remaining in the cylinder in a period from an exhaust stroke to an intake stroke, and

the controller controls the internal EGR regulator to reduce the amount of the exhaust gas remaining in the cylinder in the spark ignition mode and the transitional mode, and controls the internal EGR regulator so that the EGR ratio is the predetermined value in accordance with the amount of the exhaust gas remaining in the cylinder in the compression ignition mode.

3. The spark ignition engine of claim **2**, further comprising:

a valve mechanism configured to control operation of an intake valve and an exhaust valve, wherein

the valve mechanism constitutes the internal EGR regulator,

the valve mechanism performs a first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke in the spark ignition mode,

the valve mechanism, in addition to the first valve operation, brings the exhaust valve in an open state at timing later than timing of closing the exhaust valve in the first valve operation, or brings the intake valve in an open state at timing earlier than timing of opening the intake valve in the first valve operation in the compression ignition mode, and

the valve mechanism performs the first valve operation to operate the intake valve and the exhaust valve in the transitional mode.

4. The spark ignition engine of claim **2**, further comprising:

a valve mechanism configured to control operation of an intake valve and an exhaust valve wherein

the valve mechanism constitutes the internal EGR regulator,

in switching from the spark ignition mode to the compression ignition mode through the transitional mode,

the valve mechanism performs a first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke in the spark ignition mode,

the valve mechanism performs a second valve operation of opening the intake valve for a shorter period than a period for opening the intake valve in the first valve operation in the transitional mode, and

the valve mechanism performs, in addition to the second valve operation, a third valve operation of bringing the exhaust valve in an open state at timing later than timing of closing the exhaust valve in the second valve operation in the compression ignition mode.

5. The spark ignition engine of claim **2**, further comprising:

a valve mechanism configured to control operation of an intake valve and an exhaust valve, wherein

the valve mechanism constitutes the internal EGR regulator,

in switching from the spark ignition mode to the compression ignition mode through the transitional mode,

the valve mechanism performs a first valve operation of opening the exhaust valve in the exhaust stroke, and opening the intake valve in the intake stroke in the spark ignition mode,

the valve mechanism performs a second valve operation of opening the exhaust valve for a shorter period than a period for opening the exhaust valve in the first valve operation in the transitional mode, and

the valve mechanism performs, in addition to the second valve operation, a third valve operation of bringing the intake valve in an open state at timing earlier than timing of opening the intake valve in the second valve operation in the compression ignition mode.

6. The spark ignition engine of claim **2**, wherein

the controller switches the engine body from the compression ignition mode to spark ignition mode without intervention of the transition mode.

7. The spark ignition engine of claim **2**, further comprising:

a fuel injection valve configured to directly inject a fuel in the cylinder, wherein

the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

8. The spark ignition engine of claim **3**, further comprising:

a fuel injection valve configured to directly inject a fuel in the cylinder, wherein

the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

9. The spark ignition engine of claim **3**, wherein

the controller switches the engine body from the compression ignition mode to the spark ignition mode without intervention of the transition mode.

10. The spark ignition engine of claim **4**, wherein

the controller switches the engine body from the compression ignition mode to the spark ignition mode without intervention of the transition mode.

11. The spark ignition engine of claim **4**, further comprising:

a fuel injection valve configured to directly inject a fuel in the cylinder, wherein

the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

12. The spark ignition engine of claim **5**, wherein

the controller switches the engine body from the compression ignition mode to the spark ignition mode without intervention of the transition mode.

13. The spark ignition engine of claim **5**, further comprising:

a fuel injection valve configured to directly inject a fuel in the cylinder, wherein

the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

14. The spark ignition engine of claim **1**, wherein

the controller switches the engine body from the compression ignition mode to the spark ignition mode without switching the engine body to the transitional mode.

15. The spark ignition engine of claim **14**, further comprising:

a fuel injection valve configured to directly inject a fuel in the cylinder, wherein

the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

16. The spark ignition engine of claim **1**, further comprising:

a fuel injection valve configured to directly inject a fuel in the cylinder, wherein

the controller controls the fuel injection valve to inject the fuel at timing after an intermediate stage of a compression stroke in accordance with temperature in the cylinder after the engine body is switched from the spark ignition mode to the transitional mode.

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